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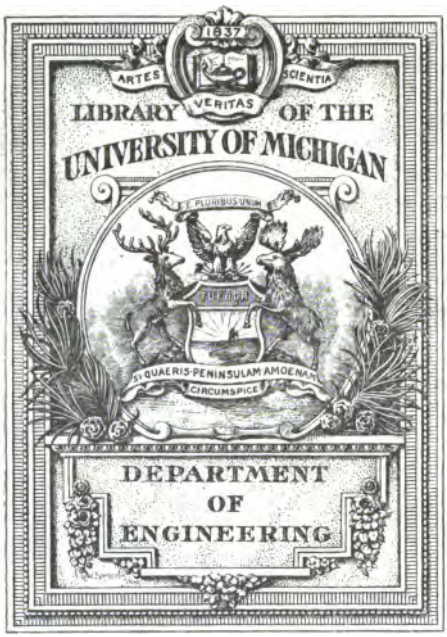
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MARINE ENGINE DESIGN

INCLUDING THE

DESIGN OF TURNING AND REVERSING ENGINES

BY

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ILLUSTRATED

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PREFACE

MOST of the text-books on the subject of marine engines deal only in a general way with the subject of design, and particularly with this subject as referring to the numerous co-ordinating parts of the modern marine engine. No matter how extensive the work, nearly all of the space is taken up with descriptions and illustrations of engines of various types, with some regard to historical features, but with very little to the actual work of laying out and designing the engine. It is felt that this book will fill a decided void in this respect and, being totally free from the descriptive part of the subject, which may be readily obtained in any text-book now on the market, it will be much more readily available for the particular use for which it is intended.

It was not the idea in this work to cover in detail the design of every part of the engine, but enough is given in the way of detailed design of the principal parts to indicate the general scope of the problem and to lay down methods by which the entire work can be carried to completion. As given, this represents the results of several years of experience in teaching the subject of marine engine design to students of the University of Michigan.

It is desired to direct attention to the desirability of systematic data-keeping in connection with the design of marine engines. Whenever possible, coefficients and factors of safety should be determined from data of similar engines working under as nearly as possible the same conditions. In this way only can it be claimed for an engine that it is DESIGNED for its work.

Chapters on the design of turning and reversing engines are included, since these engines are practically the only auxiliaries which are not usually in the hands of builders of special types. The formulæ and computations for specific cases are presented in a similar manner to those for the design of the main engine.

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MARINE ENGINE DESIGN.

The object of any heat engine is to reduce the temperature of the working fluid, this reduction of temperature being caused by the conversion of potential energy into work. In some respects the working fluid may be likened to a certain weight of water falling through a certain height. A turbine may be placed at the foot of the fall, and all of the energy of the water converted into work in one stage. A second turbine may be used half way down the fall, and the tail water from this turbine used to run the turbine at the foot of the fall; thus converting the potential energy of the water into work in two stages. In the same way, the conversion of potential energy into work may be divided into three, four, or more stages. In the heat engine, the range of temperature through which the working fluid can fall is similar to the head of water; and this range of temperature may be divided into two, three or four stages, as may seem best. When so divided, we have compound, triple, and quadruple-expansion engines. These terms apply to the number of stages employed—not to the number of cylinders.

While, in the steam engine, the reduction of temperature of the steam used is the object striven for, yet we are accustomed to deal with the temperature only indirectly. If the steam is saturated, a given temperature of steam will be accompanied by a certain pressure, so that when we reduce the pressure of the steam as low as possible, we are also reducing the temperature. In many ways it is more convenient to deal with the pressure of the steam, but the connection between pressure and temperature should not be forgotten.

A reduction in the pressure of a gas may be obtained by causing it to occupy a larger volume, and while saturated steam is not a perfect gas, yet we can reduce its pressure by causing it to occupy successively larger volumes. The steam, in expanding, will exert against the piston of the cylinder in which the expansion is effected a constantly decreasing pressure, and this pressure can be used for doing external work. If the pressure against the piston were constant, the work done would be the product of the total pressure upon the piston and the distance through which it travels; but as the pressure is constantly decreasing, we must obtain in some way the average or mean pressure acting, and multiply that by the distance moved.

If the relation between pressure and volume is one that can be expressed mathematically, such as, for instance, $P_1V_1 = PV$, we can obtain the average pressure quite easily; but, unfortunately, the relation between the pressure of the steam and the volume occupied by it as it passes through the engine cannot be expressed in any such simple way. In Fig. 1 are shown some curves giving the relation between steam pressures and volumes, obtained from the indicator cards of numerous engines. The volumes used are the total volumes in the cylinders up to cut-off, including clearance, and the pressures are the absolute pressures at cut-off. It can easily be seen that no two engines would necessarily give the same curve of relation of pressures and volumes.

Suppose that we have two engines with high-pressure cylinders of the same size, taking steam at the same pressure, and the same pressure and volume in each at cut-off. Suppose that the same work is done in each, and the mean back pressures are the same in each. The pressures at cut-off in the next, or medium-pressure, cylinders will depend upon what happens to the steam in passing from one cylinder to the other. If the relative location of cylinders is different, so that the

steam has to pass through longer pipes in one case than in the other; if the steam speeds used are different; if one pipe is better protected from radiation than the other;—the pres-

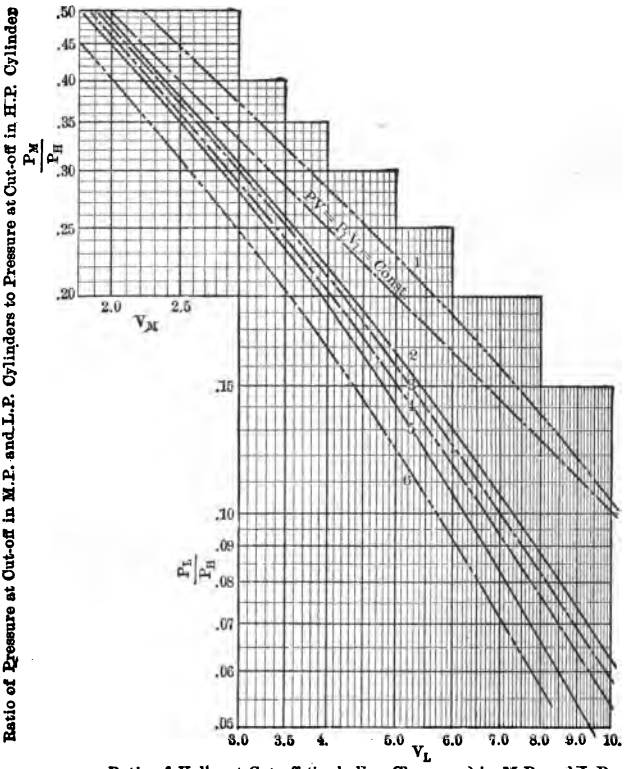


FIG. 1.

sures at cut-off will be different in the next cylinders; so we cannot obtain any one curve that will give the relation between pressure and volume for all engines.

The curves 2, 3, 4 and 5, given in Fig. 1, are for the type of

engine usual upon the ships of the merchant marine. The engine of a pumping plant, where every effort is made to get an economical engine by using jackets, reheaters, etc., will give a curve like 1; while the engine of a naval vessel may lie in the region of curve 6. These curves give the relation between pressure and volume ratios; the unit pressure and volume being that at cut-off in the high-pressure cylinder. In using such as curve for engine design it is best, wherever possible, to derive the curve from an engine similar to the one projected, but if no such data are at hand, a curve can be chosen from the region of curves 2, 3, 4 and 5, in Fig. 1. Curve 3 is for a quadruple expansion engine in which the steam speeds varied from 7,000 to 11,000 feet per minute. Curve 4 is for a naval engine, whose data are given in Cards I. and II. The steam speeds here varied from 8,000 to 13,000 feet per minute. The more efficient engines with the lower steam speeds will probably lie in the neighborhood of curves 2 and 3, while higher steam speeds and a lower efficiency will cause them to lie in the neighborhood of curves 4 and 5. The way in which such curves can be used for purposes of design will be explained later on.

CYLINDER DIMENSIONS.

There are certain data which must be available before a start can be made upon the design of any engine. It is usual to give the desired indicated horsepower, the boiler pressure, the piston speed and the number of expansions. The piston speeds for engines of the merchant marine usually range from 600 feet per minute to 1,000 feet per minute. The higher the piston speed the more attention the engine will require; the greater will be the bill for repairs, and the shorter will be the engine's life. The number of expansions to be used for a given boiler pressure will be determined by experience, and will be a

compromise between the number which will give the greatest economy and that which will give the engine of least weight.

The formula for indicated horsepower is

$$I.H.P. = \frac{2pLan}{33{,}000}, \qquad (1)$$

where p is the mean unit pressure acting upon an area a, through a distance of $2ln$, or the piston speed; l is the length of stroke in feet, and n is the revolutions per minute. Referring to the data supposed to be available, it will be seen that the unknowns are

$$pa = \frac{I.H.P. \times 33{,}000}{\text{piston speed}} \qquad (2)$$

As stated before, work is derived from the steam by reducing its temperature by means of expansion; the increase in volume and decrease in temperature being accompanied by a decrease in pressure. The actual rate at which the pressure decreases cannot be stated in any simple form, but it is usual and convenient to assume the relation between pressure and volume to be given by the formula $P_1 V_1 = PV = \text{constant}$; the character of the curve being that shown in Figs. 1 and 2. The deviation of the actual rate from the assumed rate, and certain losses, are allowed for by using a design factor derived from engines in use. The design factor F for the engine whose data are given in Cards I. and II., is 0.577, and is obtained by dividing the area inclosed in the dotted lines of Fig. 2 by the area *aturx*.

The mean pressure exerted upon an unit area while the steam is expanding from an initial pressure P_1 and volume at, to the volume wv, will be given by the formula

$$MP = P_1 \frac{(1 + \log_e R)}{R}, \qquad (3)$$

where R is the number of expansions

$$= \frac{wv}{at} = \frac{\text{final volume}}{\text{initial volume}}.$$

The proof of this can be found in any book on thermodynamics. The theoretical mean effective pressure will be

$$T.M.E.P. = P_1 \frac{(1 + \log_e R)}{R} - P_b, \tag{4}$$

and the actual mean effective pressure to be expected in the engine will be

$$MEP = \left[P_1 \frac{(1 + \log_e R)}{R} - P_b \right] F. \tag{5}$$

This MEP in (5) will be the p of (2), so that the value of the only remaining unknown a can be found. This a will be the area of the cylinder of the single-stage expansion engine, or the area of the low-pressure cylinder of any engine where the expansion or reduction of temperature is effected in more than one stage. In order to see this, it is only necessary to remember that in order to produce the desired indicated horsepower, starting with steam of a given pressure and expanding it a certain number of times, a certain weight or volume of steam will have to be used. After this steam has been expanded the chosen number of times, it will occupy a definite volume, irrespective of the number of stages of expansion, and that final volume will be the volume of the low-pressure cylinder. Therefore,

$$LP = a = \frac{I.H.P. \times 33{,}000}{P.S. \times MEP}. \tag{6}$$

A low-pressure cylinder having been provided, of sufficient capacity to allow for the desired expansion of the weight of steam necessary to produce the desired indicated horsepower, the next step is to provide a cylinder of sufficient size to take

in the necessary amount of steam from the boiler. As it is desirable that some portion of the total expansion should take

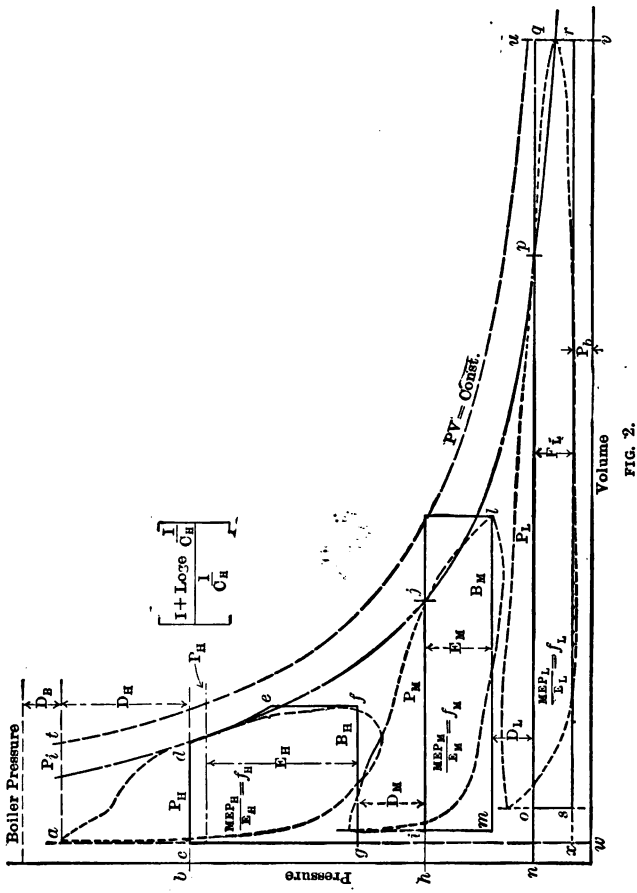


FIG. 2.

place in this cylinder, it will not take steam from the boiler during its full stroke, but will cut off communication at some fraction of the stroke. The number of expansions

$$R = \frac{\text{final volume of steam}}{\text{initial volume of steam}} =$$

$$\frac{\text{area of L. P. cyl.} \times \text{stroke}}{\text{area of H. P. cyl.} \times \text{stroke} \times \text{fraction of cut-off}} = \frac{LP}{HP \times C_h}.$$

Therefore,

$$HP = \frac{LP}{R \times C_h}. \qquad (7)$$

If the expansion takes place in one cylinder, or in a single-stage engine, then

$$HP = LP, \text{ and } C_h = \frac{LP}{HP \times R} = \frac{1}{R};$$

and the cut-off fraction will be the reciprocal of the number of expansions.

Between the high-pressure cylinder and the low-pressure cylinder we can put as many cylinders as is thought advisable. The number will not affect materially the total power developed, but will merely divide the expansion into as many stages as is necessary for economical use of the steam. The sizes of these intermediate cylinders, and the cut-offs in all cylinders except the high-pressure cylinder, will affect only the distribution of work among the cylinders.

The reason for using two or more stages for the expansion of the steam is to avoid the loss by initial condensation that occurs when the range of temperature in one cylinder is more than 75 or 80 degrees F. In a double-acting engine, at the beginning of the stroke, there is upon one side of the piston, steam at the temperature of the source of supply, and on the other, steam at the temperature of exhaust. As the piston moves, the high-temperature steam comes in contact with a portion of the cylinder wall which just previously has been in contact with the low-temperature steam, and condensation

results. It is to reduce this difference of temperature, and the consequent condensation, that the range of temperature in one cylinder is limited by dividing the total expansion into two or more stages. Thus the number of stages will depend upon the total range of temperature, or upon the initial pressure of the steam to be used. In Table I. are shown the values of the different quantities for merchant engines and naval engines.

TABLE I.

Type.	Initial Pressure at Engine, Absolute. P_i	Nominal Expansions. R	High-pressure Cutoff. C_h	Back Pressure in Low-pressure Cylinder. P_b	Design Factor. F
Triple naval.....	165 to 200	6 to 8½	0.70 to 0.80	5 to 7	0.58 to 0.70
Triple merchant.....	165 to 185	9 to 12	0.65 to 0.75	5 to 6	0.65 to 0.75
Quadruple merchant...	190 to 210	12 to 14	0.65 to 0.75	5 to 6	0.55 to 0.70

Collecting together the principal formulæ used so far, we have:

$$MEP = \left[P_i \frac{(1 + \log_e R)}{R} - P_b \right] F \qquad (5)$$

$$\text{Area of } LP = \frac{I.H.P. \times 33{,}000}{PS \times MEP} \qquad (6)$$

$$\text{Area of } HP = \frac{LP}{R \times C_h} \qquad (7)$$

The stroke has usually some one of the following values: 18, 21, 24, 27, 30, 33, 36, 39, 42, 45, 48, 54, 60, 66, or 72 inches. The ordinary merchant engine will have one of the above values of stroke, nearest the square root of the product of the diameters of the high-pressure and low-pressure cylinders. In

naval engines the stroke will be less than this, in extreme cases being only 50 percent of the above value. On the other hand, in slow-running, economical freight steamer engines, it will be greater by 10 to 15 percent.

The proper diameter for the intermediate-pressure cylinder or cylinders, and the cut-off in the mean-pressure and low-

CARD I.

Name of Ship.	H.P.		1 M.P.		2 M.P.		L.P.	
Size of engine $\frac{27—41—64}{30}$	top	bot.	top	bot.	top	bot.	top	bot.
Cutoff—Designed or apparent,	.77	.71	.78	.68			.75	.69
Cutoff—Mean,	.74		.73				.72	
Clearances,								
Clearances—Mean of top and bottom,	.134		.10				.069	
Initial pressure—Absolute,	163	163	78	76			27	29
Cutoff pressure—Absolute,	123	127	49	54			17¼	19¼
Cutoff pressure—Mean absolute,	125		51.5				18.5	
Average back pressure—absolute,	73		31				6.5	
Effective cutoff pressure,			20.5				12.0	
M. E. P. from indicator cards,	53.5	59.2	23.5	25.4			14.0	13.4
Mean of top and bottom, M. E. P.,	56.4		24.4				13.7	
I. H. P. developed in cyl. P. S. = 838	824		820				1115	
M. E. P. when power is equally distrib.,	63.3		27.4				11.26	

pressure cylinders, will be chosen to give a good distribution of work among the cylinders. In order to make this choice with intelligence, the data obtained from engines in service should be worked up in such a way as to be readily used. A system of data keeping which has been found to give good results deals with the following quantities and relations: The curve of relation between cut-off pressures and total volumes at cut-off in the different cylinders (see curve *djp*, Fig. 2); the drop D_h from the initial pressure to the cut-off pressure in the

high-pressure cylinder; the mean back pressure in each cylinder, B_h, B_m, P_b; the drop from the mean back pressure of one cylinder to the cut-off pressure of the next succeeding, D_m and D_l; the relation between the mean effective pressure of the high-pressure indicator card MEP_h and the mean effective pressure E_h of a card *cdefg*, which assumes that the steam

CARD II.

No. of expansions $= \left[\frac{64}{27}\right]^2 \frac{1}{0.74} = 7.61$

$163 - \frac{3.0295}{7.61} - 6.5 = 65 - 6.5 = 58.5$

$56.4\left[\frac{27}{64}\right]^2 = 10.05$

$24.4\left[\frac{41}{64}\right]^2 = 10.02$

13.7

33.77

$\frac{33.77}{58.5} = 0.577 = \text{Design factor } F.$

$D_h = 163 - 125 = 38$

$D_m = 73 - 51.5 = 21.5$

$D_l = 31 - 18.5 = 12.5$

$72.0 = D$

$\frac{72}{163} = 0.442$

$125\frac{1 + 0.3001}{1.35} = 120$ $\quad 73$ $\quad 47$

$\frac{56.4}{47} = 1.20 = f_h$

$\frac{24.4}{20.5} = 1.19 = f_m$

$\frac{13.7}{12} = 1.14 = f_l$

For power equally distributed

$\frac{33.77}{3}\left[\frac{64}{27}\right]^2 = \frac{63.3}{56.4}$ $\quad 6.9$ $\quad \frac{63.3}{47+6.9} = 1.17 = f_h$

$\frac{33.77}{3} \times \left[\frac{64}{41}\right]^2 = \frac{27.4}{24.4}$ $\quad 3.$ $\quad \frac{27.4}{20.5+3} = 1.165 = f_m$

$11.26 - 13.7 = -2.4$

$\frac{11.26}{12-2.4} = 1.172 = f_l$

$\frac{51.5}{125.} = 0.412 = P_m$ $\qquad$ $\frac{18.5}{125.} = 0.148 = P_l$

$\frac{(41)^2 \times (0.73 + 0.10)}{(27)^2 \times (0.74 + 0.134)} = 2.19 = V_m$

$\frac{(64)^2 \times (0.72 + 0.069)}{(27)^2 \times (0.74 + 0.134)} = 5.07 = V_l$

enters at the cut-off pressure and expands with PV = constant,

$$\frac{MEP_h}{E_h} = f_h;$$

the relation between the MEP of the mean-pressure and low-pressure cylinder indicator cards and the effective cut-off pressures in those cylinders,

$$\frac{MEP_m}{E_m} = f_m, \quad \frac{MEP_l}{E_l} = f_l.$$

This data can be kept on a card similar to that shown in Card I., and when worked up, as shown in Card II. on the back of the card, the necessary factors and relations will be available.

There are various ways of getting the design factor F. In this system of data keeping the engine is not charged with the loss of pressure that occurs between the boiler and the engine, as this will vary with the relative location of engine and boilers, steam speeds used and general layout of piping. The values given for F, in Table I., consider the steam as entering the engine with the initial pressure shown on the high-pressure indicator card, expanding the nominal number of times with $P_1V_1 = PV =$ constant, and with a back pressure equal to the mean back pressure of the low-pressure cylinder indicator card. The nominal number of expansions does not take into consideration the reduction of expansion due to clearances.

The design factor F takes account not only of the deviation of the actual relation of pressure and volume from the assumed, but also of the losses of pressure that occur while the steam is passing from one cylinder to the next. We should expect, therefore, that the sum of the drop $D = D_h + D_m + D_l$ would depend upon the value of the design factor. It has been found that there is such a relation, and the approximate values of D are given in connection with F, as a portion of the absolute initial pressure P_1.

TABLE II.

F =	0.6	0.65	0.675	0.70	0.725	0.75	0.80
$\frac{D}{P_i}$ =	.35—.40	.25—.29	.235—.27	.225—.25	.21—.23	.20—.215	.175—.185

D_h = 0.5 to 0.6D; D_m = 0.25 to 0.35D; D_l = 0.12 to 0.17D

The drop D_b from boiler to engine will vary from 10 to 25 pounds, usually being from 15 to 20 pounds. The relative values of D_h, D_m and D_l will not be constant, as can be easily seen when the causes of the drop are considered. No two engines will necessarily work under the same conditions, so that the relative values can be only approximately indicated. Whenever possible, the various factors, drops, etc., should be derived from engines similar to the one projected, working under like conditions.

The card factors f_h, f_m and f_l should be reduced to some standard condition. The character of the admission line and back pressure line of an indicator card will not change essentially if there is 10 percent more or 10 percent less work developed in the cylinder. The two lines will be bodily separated or brought closer together. The standard condition chosen in the case of the triple engine is that which exists when one-third of the total indicated horsepower is developed in each cylinder. Thus, if the *MEP* of an indicator card is 31, and the effective cut-off pressure is 28, while the *MEP* should be 33 in order that one-third of the work may be done in that cylinder, the corrected value of f_m is

$$\frac{31 + 2}{28 + 2} = 1.1.$$

It can be seen by referring to Fig. 2 that f_h will vary with the drop D_h, and f_m and f_l will vary with the fraction of cut-off in the cylinder. It was found that the values of f_h, f_m and f_l, when reduced to the standard condition, could be given by the formulæ,

$$f_h = 1 + \frac{D_h - 16}{100} \tag{8}$$

$$f_m = 2.25\ C_m - 0.475 \tag{9}$$

$$f_l = 1.25\ C_l + 0.256 \tag{10}$$

where D_h is the drop from initial to cut-off pressure, and C_m

and C_1 are the fractional cut-offs in the mean-pressure and low-pressure cylinders. The way in which these factors and relations of pressure and volume can be used for design purposes can be best shown by means of an example:

I. H. P. = 3,000; *P. S.* = piston speed = 850 feet per minute; boiler pressure = 185 pounds gage; P_b = back pressure in low-pressure cylinder = 5.5 pounds; R = number of expansions = 11; C_h = cut-off in high-pressure cylinder = 0.675; F = design factor = 0.7. Work to be equally divided among the three cylinders.

The initial pressure at the engine will be taken as 170 pounds gage, or 185 pounds absolute, allowing a drop of 15 pounds from the boiler to the engine. Table II. gives the allowable total drop for a design factor of 0.7 as 0.225 to 0.25 of P_1. We will use $D = 0.24\ P_1 = 0.24 \times 185 = 44.5$ pounds.

$$D_h = 44.5 \times 0.55 = 24.5; \quad D_m = 44.5 \times 0.30 = 13.5;$$
$$D_l = 44.5 \times 0.15 = 6.5$$

From (5) $$MEP = \left[185\frac{1 + 2.398}{11} - 5.5\right] \times 0.7$$
$$\left[= 57.1 - 5.5\right] \times 0.7 = 36.1$$

From (6) $$LP = \frac{3{,}000 \times 33{,}000}{850 \times 36{,}1} = 3{,}230 \text{ square inches; diameter} = 64 \text{ inches.}$$

From (7) $$HP = \frac{3{,}230}{11 \times 0.675} = 435 \text{ square inches; diameter} = 23\tfrac{1}{2} \text{ inches.}$$

$$\sqrt{64 \times 23\tfrac{1}{2}} = 38.8. \quad \text{Let stroke} = 42 \text{ inches.}$$

In order that one-third of the total work may be done in the high-pressure cylinder, the *MEP* of the indicator card must be

$$MEP_h = \frac{36.1}{3} \times \left[\frac{64}{23.5}\right]^2 = 89 \text{ pounds.}$$

From (8) $f_h = 1 + \frac{24.5 - 16}{100} = 1.085$; $E_h = \frac{89}{1.085} = 82$ pounds

$P_h = 185 - 24.5 = 160.5$ pounds.

$$B_h = P_h \frac{1 + \log_e \frac{1}{C_h}}{\frac{1}{C_h}} - E_h = 160.5 \times \frac{1+0.392}{1.48} - 82 = 152 - 82 = 70$$

$P_m = B_h - D_m = 70 - 13.5 = 56.5$ pounds.

$\frac{P_m}{P_h} = \frac{56.5}{160.5} = 0.352.$

The volume ratio that will give this pressure ratio, 0.352, will be found by using curve 3 in Fig. 1. The volume ratio $V_m = 2.63$. These values of P_m and V_m will be used later. Having found P_m, we will now proceed to get B_m by working up from the low-pressure cylinder.

We have assumed that curve 3, Fig. 1, gives the relation between pressure ratios and volume ratios for the engine we are designing, so that when we assume a certain cut-off in the low-pressure cylinder the cut-off pressure is determined from the curve. This cut-off pressure must be sufficient to give an effective cut-off pressure E_1, of such a value as will insure that one-third of the work is done in that cylinder. The relation between E_1 and MEP_1 will be given by (10).

In order to get the volumes up to cut-off, we must know the clearances. It will be best to determine these by comparison with a similar engine, but if no such data are at hand the following clearances can be assumed for merchant ship engines, where the stroke $= \sqrt{\text{high-pressure diam.} \times \text{low-pressure diam.}}$:

High-pressure cylinder	0.17 to 0.20.
Medium-pressure cylinder	0.13 (piston valves); 0.11 (slide valves).
Low-pressure cylinder	0.12 (piston valves); 0.09 (slide valves).

These values will increase as the stroke gets relatively shorter, so that they should be mutiplied by

$$\sqrt{\frac{\text{High-pressure diameter} \times \text{low-pressure diameter}}{\text{stroke}}}.$$

to get the approximate clearances in other cases.

We will take the clearances as $Cl_h = 0.17$, $Cl_m = 0.13$, $Cl_l = 0.12$ for the high-pressure, mean-pressure and low-pressure cylinders, respectively,

The value of P_l can be obtained from curve 3, Fig. 1, as follows: The volume ratio will be

$$V_l = \frac{(C_l + Cl_l)LP \times s}{(C_h + Cl_h)HP \times s}. \qquad (11)$$

With this value of V_l the value of P_l/P_h can be obtained from the curve, and knowing P_h, the value of P_l is known. The value P_l must also be such as will cause the necessary work to be done in the cylinder, or,

$$P_l = P_b + \frac{MEP_l}{1.25C_l + 0.256} \qquad (12)$$

In order to find the value of P_l which will satisfy both equations (11) and (12), we will assume values of C_l, plot the resulting values of P_l on C_l, and find where the two curves cross.

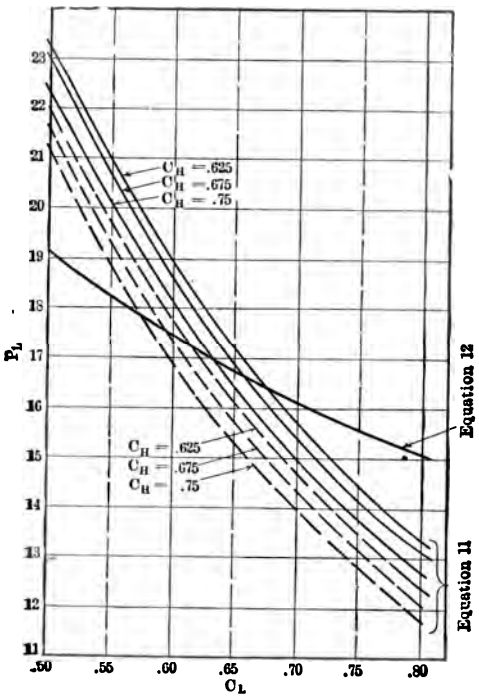
From (11) $V_l = \dfrac{(C_l + 0.12)3{,}230}{(0.675 + 0.17)435} = (C_l + 0.12) \times 8.8$

TABLE III.

C_l	V_l	From Fig. I. P_l/P_h	P_l
0.5	5.45	0.144	23.1
0.6	6.33	0.116	18.6
0.7	7.22	0.096	15.4
0.8	8.08	0.082	13.2

From (12) $$P = 5.5 + \frac{12.03}{1.25\,C_1 + 0.256}$$

C_1	P_1
0.5	19.17
0.6	17.46
0.7	16.17
0.8	15.1



Full Lines from Curve 3, Fig. 1.
Broken " " " 4, Fig. 1.

FIG. 3.

When these values are plotted upon C_1 as abscissæ, the resulting curves cross at a pressure of 16.65 pounds, and with $C_1 = 0.655$ (see Fig. 3), $C_h = 0.675$.

$$B_m = P_l + D_l = 16.65 + 6.5 = 23.15$$

The value of $P_m = 56.5$ has been previously obtained.

$$E_m = P_m - B_m = 56.5 - 23.15 = 33.35$$

We are now ready to determine the mean-pressure diameter and cut-off. Here again there are two conditions which must be fulfilled, and consequently two equations to use. The value of E_m, when multiplied by the factor f_m, depending upon the cut-off, must give a MEP_m that will do the necessary work when acting upon the mean-pressure cylinder area. This gives the equation:

$$E_m \times f_m = MEP_m = \frac{MEP \times LP}{3 \times MP} = E_m(2.25C_m - 0.475)$$

or
$$MP = \frac{MEP \times LP}{3E_m(2.25C_m - 0.475)}. \qquad (13)$$

This area of mean-pressure cylinder and cut-off must also be such as will give the proper volume ratio V_m previously obtained from the pressure ratio P_m/P_h.

$$\frac{MP(C_m + Cl_m) \times s}{HP(C_h + Cl_h) \times s} = V_m$$

or,
$$MP = \frac{V_m \times HP(C_h + Cl_h)}{C_m + Cl_m} \qquad (14)$$

We can equate (14) and (13) and solve for C_m, the only unknown.

Let $X = V_m \times HP(C_h + Cl_h)3E_m$, and
$Y = MEP \times LP$.

Then,
$$CM = \frac{0.475X + Cl_m Y}{2.25X - Y}.$$

This value of C_m, substituted in either (13) or (14), will give the area of the mean-pressure cylinder. For the engine we are designing.

$$X = 2.63 \times 435(0.675 + 0.17) \times 3 \times 33.35 = 96{,}200.$$
$$Y = 36.1 \times 3{,}230 = 116{,}700.$$
$$C_m = \frac{0.475 \times 96{,}200 + 0.13 \times 116{,}700}{2.25 \times 96{,}200 - 116{,}700}$$
$$= \frac{45{,}700 + 15{,}150}{217{,}000 - 116{,}700} = \frac{60{,}050}{101{,}300} = 0.60$$
$$MP = \frac{2.63 \times 435(0.675 + 0.17)}{0.60 + 0.13} = 1{,}328, \text{ diameter} = 41 \text{ inches.}$$

Collecting together the quantities calculated and assumed so far, we have

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches.....	23½	41	64
Cut-off.....	0.675	0.60	0.655
Clearance.....	0.17	0.13	0.12
Stroke, inches.....	42	42	42

If the high-pressure cut-off had been taken as 0.625, the results would have come as follows (see Fig. 3, full line curve, $C_h = 0.625$):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches.....	24½	42	64
Cut-off.....	0.625	0.56	0.67

If the high-pressure cut-off had been taken as 0.75, the following results would have been obtained (see Fig. 3, full line curve, $C_h = 0.75$):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches.....	22½	39½	64
Cut-off.....	0.75	0.71	0.635

In computing the foregoing results, curve 3 in Fig. 1 was used. If curve 4 had been used, the results would have been as follows (see Fig. 3, broken curves):

	High-Pressure.	Medium-Pressure.	Low-Pressure.
Diameter in inches.....	23½	39	64
Cut off.....	0.675	0.655	0.595
Diameter in inches.....	24½	40	64
Cut off.....	0.625	0.61	0.615
Diameter in inches.....	22½	36¾	64
Cut off.....	0.75	0.81	0.575

It may not be thought advisable to use any one of the sets of results as it stands. The effect of any change, however, can be foreseen, as an increase of the cut-off in any cylinder will cause less power to be developed there and more power in the preceding cylinder. A shortening of the cut-off will have the reverse effect. It will be noticed that an increase of the high-pressure cut-off is accompanied by an increase of the mean-pressure cut-off, and a decrease of the low-pressure cut-off if equal powers are to be developed in each cylinder; while a decrease in the high pressure cut-off causes the mean pressure to decrease and the low pressure to increase.

STRENGTH OF MATERIALS.

The cylinder diameters and cut-offs having been determined, we are in a good position to approximate to the loads coming upon the different parts, and to proportion them so that they shall be strong enough to carry these loads. There are a great many formulæ in use for this purpose, but many of them are in such shape that they cannot be readily adapted to all conditions. Most of them are based upon the assumption that such a certain ultimate strength is to be used, and that, therefore, the allowable working stress is a fixed quantity. Since, however, the ultimate strength of the steel used in marine engines varies from 50,000 pounds to 90,000 pounds per square inch, it is much better to base the calculations upon the allowable factor of safety, and that is the principle which will be followed in this section of design.

The factor of safety which it is allowable to use in the design of a given part depends upon the kind of load to which it is subjected, the construction of the part itself, and upon the conditions under which it works. There are three kinds of loads: steady loads, intermittent loads and alternating loads. The steady load is one which is applied originally in an appreciable length of time, and the stress resulting in either tension or compression of constant amount. The intermittent load is one which is applied more or less suddenly, and produces either tension or compression; the stress varying from zero to a maximum, or from a minimum to a maximum. The alternating load is of such a nature as to cause the stress to change alternately from tension to compression.

The elastic limit is the limit of the stress to which a material may be subjected, and have the strain or deformation proportional to the stress. The elastic limit is not the same under all conditions. When the steel comes from the mill, it may have an unnatural elastic limit, due to its treatment during manufacture. When subjected to a moderate alternating load for a short time, the effect of this treatment will be overcome, and the piece will have more nearly its natural elastic limit. The tension elastic limit may be increased by subjecting it to successively increasing loads, but its elastic limit in compression will be correspondingly decreased; so that a piece which has been subjected to large intermittent tension loads might fail at a comparatively small load in compression. The elastic limit in compression can be restored by subjecting the piece to successively increasing compression loads, the elastic limit in tension then being correspondingly decreased.

It has been found by experiment that a piece which would carry a constant total load of 80,000 pounds tension for an indefinite time, would fail after a while if the load were an intermittent one varying from 80,000 pounds to 40,000 pounds; it would fail sooner if the load varied from 80,000 pounds to

zero; and would fail in a comparatively short time if subjected to an alternating load varying from 80,000 pounds in tension to 80,000 pounds in compression. In order to have the piece equally strong under each condition of loading, the maximum of 80,000 pounds constant load which the piece will carry indefinitely should be reduced to 40,000 pounds if the load is intermittent, and varies from 40,000 pounds to zero, and to something between 40,000 pounds and 80,000 pounds if the minimum is 30,000 pounds instead of zero. If the stress varies from tension to compression, the load should vary from not more than 27,000 pounds tension to 27,000 pounds compression, in order that the piece may last indefinitely. In other words, the destructiveness of the three kinds of loads, constant, intermittent and alternating, is about in the ratio of 1, 2, 3, respectively.

Experiment has shown that for steel the elastic limit is from one-half to two-thirds of the ultimate strength. It is customary to choose the working stress so that it shall be less than the elastic limit of the material, usually less than one-half of it. It results, therefore, that in some structural work on land, the factors of safety employed are 3, 6 and 9 for steady, intermittent and alternating loads.

The statement was made above that the conditions under which the parts work affect the factors of safety. The breaking of the parts of an engine in a ship would be attended with so much danger, and the facilities for repair are so limited, that somewhat larger factors of safety should be employed for marine engines than for structural work on land. In this system of design, the factors of safety used for engines of the merchant marine are 4, 8 and 12 for steady, intermittent and alternating loads, respectively.

In engines for the vessels of the navy, the factors of safety employed are more nearly 3, 6 and 9. The reasons for this are that it is desirable to save weight, the life of the engine is

comparatively short, and the engines are seldom run at full power. Naval vessels are usually cruising around at one-half or two-fifths power, so that although the factors of safety are smaller for full power, they are amply large for the reduced power which the engines develop during the greater part of their existence.

Practically there are no parts of a marine engine which can be designed for steady loads. Some of the bolts and threaded

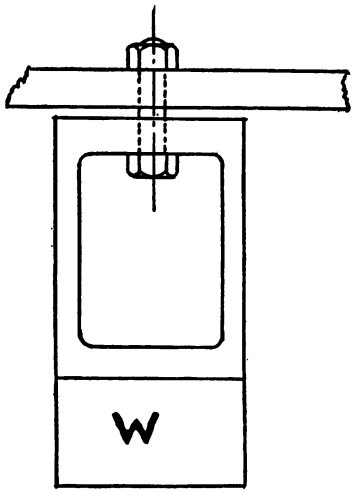


FIG. 4.

parts may be subjected to steady loads if the nuts are set up tight enough, but it would not be safe to count on the nuts always being as tight as when they were first set up. The stress in the body of a bolt is always that due to setting up the nut, unless the load upon the bolt is in excess of this. Referring to Fig. 4, it can be seen that the stress in the body of the bolt will be that due to the load W as long as the flanges are apart. When the nut is set up sufficiently to bring the flanges together the stress will be increased, and the body of

the bolt will not be stressed further until sufficient weight has been added to W to cause the flanges to separate again. In this way, even with an intermittent load acting upon the parts joined by the bolt, the body of the bolt might be subjected to a steady load if the nut were set up tight enough. in the first place, to subject the bolt to a stress greater than the intermittent load. It would not be advisable to count on this, however, so that a factor of safety as low as 4 should never be used in any part of a marine engine. The lowest factor of safety that we shall use will be 8 for merchant engines and 6 for naval engines.

The parts subjected to intermittent loads are bolts, the threaded portion of piston rods and certain portions of valve stems. In the case of bolts and other threaded parts, a deviation from the chosen factor of safety 8 is made, because of the construction of the part. A piece of steel which has been nicked will fail at a lower stress, the net area at the nick being used, than if the same net area were in a piece without a nick. The sharp angle at the root of the threads has the same effect upon the bolt as the nick just referred to, so that the metal in the threaded portion cannot be stressed as greatly as it could be if the surface of the bolt were perfectly plain. The factor of safety to be used for bolts will be increased on this account to 10 for merchant engines and to 8 for naval engines.

Another condition which causes a further modification of the factor of safety for bolts is the initial torsional stress set up when the nut is put on, due to the friction between the threads of the nut and bolt. This initial stress is greater relatively in bolts of small diameter than in bolts of larger diameter, and can be considered as negligible in bolts of 3 inches diameter and above; so that, starting with a factor of safety of 10 for bolts of 3 inches diameter and above, it will be gradually increased until it is 16 for bolts of 1 inch diameter. In bolts for naval engines, the factor of safety will vary from

8 for bolts of 3 inches diameter and above to 14 for 1-inch bolts. Below is given a table of the factors of safety, areas and allowable loads upon bolts of various diameters, upon the assumption that steel of 60,000 pounds per square inch ultimate strength is used. For steel of higher strength, the allowable load is increased in proportion. It is customary to use not less

TABLE IV.

Diameter of Bolt.	No. of Threads per Inch.	Area at Root of Thread.	Merchant.		Naval.	
			Factor of Safety.	Allowable Load in Pounds.	Factor of Safety.	Allowable Load in Pounds.
3/4	10	.302	16.70	1,090	14.70	1,240
7/8	9	.420	16.40	1,540	14.40	1,755
1	8	.550	16.00	2,060	14.00	2,355
1 1/8	7	.694	15.60	2,670	13.60	3,065
1 1/4	7	.893	15.25	3,520	13.25	4,060
1 3/8	6	1.057	14.90	4,250	12.90	4,910
1 1/2	6	1.295	14.50	5,360	12.50	6,225
1 5/8	5 1/2	1.515	14.10	6,450	12.10	7,525
1 3/4	5	1.746	13.75	7,625	11.75	8,920
1 7/8	5	2.051	13.40	9,170	11.40	10,800
2	4 1/2	2.302	13.00	10,050	11.00	12,600
2 1/4	4	3.023	12.25	14,800	10.25	17,700
2 1/2	4	3.719	11.50	19,400	9.50	23,500
2 3/4	4	4.62	10.75	25,700	8.75	31,600
3	4	5.63	10.00	33,800	8.00	42,250
3 1/4	4	6.73	10.00	40,400	8.00	50,500
3 1/2	4	7.90	10.00	47,400	8.00	59,250
3 3/4	4	9.21	10.00	55,200	8.00	69,000
4	4	10.60	10.00	63,500	8.00	79,350
4 1/4	4	12.10	10.00	72,500	8.00	90,700
4 1/2	4	13.68	10.00	82,100	8.00	102,800
4 3/4	4	15.36	10.00	92,100	8.00	115,200
5	4	17.20	10.00	103,000	8.00	129,000
5 1/2	4	21.05	10.00	126,400	8.00	157,500
6	4	25.30	10.00	152,000	8.00	190,000
6 1/2	4	30.00	10.00	180,000	8.00	225,000
7	4	35.00	10.00	210,000	8.00	262,500

than 4 threads per inch for the bolts of connecting rods, caps and main bearings and for the threaded portion of the piston rod.

When hexagonal nuts and heads are used, they are usually of standard dimensions, but a great many cylindrical nuts and heads and collar nuts are used. For cylindrical nuts, $d = 1.75$ D to 1.67 D, and $h = D$, where d = diameter of the nut, and h its height. D = diameter of the bolt.

For cylindrical bolt heads: $d = 1.5\ D$; $h = 0.67\ D$.

The ultimate strength of the materials used in marine engines will be assumed as follows:

Cast iron.....	20,000 pounds (tension).
Cast steel.....	55,000 to 70,000 pounds.
Wrought steel.....	60,000 to 100,000 pounds.
Phosphor bronze.....	35,000 pounds.
Manganese bronze.....	50,000 pounds.

We give a table of collar nuts, where the flange of the part joined is counterbored to serve as a collar for the nut. Sometimes the collar is a separate piece attached to the flange by pins or dowels.

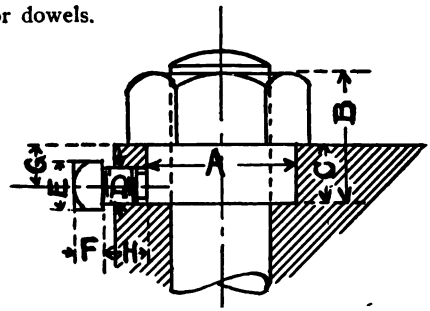


FIG. 5. DIMENSIONS IN INCHES.

TABLE V.

Diameter.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H**	*H*†
1/2	13/16	7/8	7/16	1/4	3/8	1/4	5/16	5/16	3/8
9/16	7/8	7/8	7/16	1/4	3/8	1/4	5/16	5/16	3/8
5/8	1	1	7/16	1/4	3/8	1/4	5/16	5/16	3/8
3/4	1 3/16	1 1/8	9/16	5/16	7/16	5/16	3/8	3/8	7/16
7/8	1 3/8	1 1/4	9/16	5/16	7/16	5/16	3/8	3/8	7/16
1	1 9/16	1 3/8	5/8	3/8	1/2	5/16	7/16	7/16	1/2
1 1/8	1 3/4	1 3/8	5/8	3/8	1/2	5/16	7/16	7/16	1/2
1 1/4	1 15/16	1 5/8	3/4	7/16	5/8	3/8	1/2	1/2	5/8
1 3/8	2 1/8	1 3/4	3/4	7/16	5/8	3/8	1/2	1/2	5/8
1 1/2	2 5/16	2	3/4	7/16	5/8	3/8	1/2	1/2	5/8
1 5/8	2 1/2	2 1/8	7/8	1/2	11/16	7/16	9/16	1/2	5/8
1 3/4	2 11/16	2 1/4	7/8	1/2	11/16	7/16	9/16	1/2	5/8
1 7/8	2 7/8	2 3/8	7/8	1/2	11/16	7/16	9/16	1/2	5/8
2	3 1/16	2 1/2	7/8	1/2	11/16	7/16	9/16	1/2	5/8
2 1/2	3 3/4	2 7/8	1	5/8	13/16	1/2	11/16	5/8	3/4
3	4 1/2	3 1/8	1	5/8	13/16	1/2	11/16	5/8	3/4
4	6	4	1 1/4	3/4	1	5/8	7/8	3/4	1
5	7 1/2	5	1 1/2	7/8	1 1/8	3/4	1	7/8	1 1/8
6	9	6	1 3/4	1	1 1/4	13/16	1 1/8	1	1 1/8

* Wrought iron and composition. † Cast iron.

CYLINDERS.

The cylinders may be divided into two classes, those with liners and those without liners. If the cylinders are to be jacketed, of course liners will be necessary, but very often liners are used when it is not intended that the space between the liner and the cylinder barrel shall be used for jacket steam. The presence of a liner makes it easier to warm up the engine before starting, by temporarily admitting steam to the jacket space. The liners are very simple in shape, and can be made of a tough grade of cast iron, which will give a good wearing surface for the piston rings, while the more complicated cylinder casting can be made of a softer grade, which will flow readily during casting. A further advantage in the use of a liner is that, if any accident happens to the piston, it may break the liner, which can be readily replaced, rather than the barrel of the cylinder, the replacing of which would necessitate the dismantling of a good part of the engine.

In deciding upon whether or not steam jackets are to be used, the following points should be considered: range of temperature in one cylinder, temperature of steam to be used in jacket spaces and number of revolutions of cranks per minute. The layer of metal in the cylinder walls which is subjected to any great variation of temperature is usually comparatively thin, and the purpose of the steam jacket is to keep this layer as thin as possible. The layers of metal near the outside of the wall are almost constant in temperature, and the temperature of the jacket steam must be considerably higher than the natural temperature of the outside of the wall, in order that its effect may be felt upon the inner layers, and result in any reduction of the amount of metal subjected to variations of temperature. The use of boiler-pressure steam in the jackets of the high-pressure cylinder of triple and quadruple expansion engines has, therefore, very little beneficial effect upon the economy of the engine; and, if the

space is not well drained of water, may have a harmful effect. It is quite common to use no steam in the high-pressure jackets, but to use jacket steam on the medium-pressure and low-pressure cylinders only. The element of time will also affect the thickness of the layer of metal subjected to variation of temperature, so that the higher the number of revolutions per minute the less need there is of jacket steam.

The load upon the cylinder walls of the high-pressure and medium-pressure cylinders is intermittent in character, varying from a minimum to a maximum. In the low-pressure cylinder, when a condenser is used, the pressure at times will be less than atmospheric, and consequently the load will be alternating in character. The load to be used in calculating the thickness of the walls will depend upon whether or not jacket steam is to be used. If the jackets are not to be used except for warming up, and reducing valves and relief valves are fitted, so that the pressure in the jackets cannot be greater than that for which they are designed, it may be assumed that the load upon the cylinder walls is as follows:

High-pressure — maximum = boiler pressure, gage.
Triple; medium-pressure — maximum = 0.5 boiler pressure.
Quadruple; first medium-pressure — maximum = 0.6 boiler pressure.
Quadruple; second medium-pressure — maximum = 0.4 boiler pressure.
Quadruple and triple; low-pressure — maximum = 0.25 boiler pressure.

The maximum pressure in the low-pressure cylinder will seldom be more than 35 pounds absolute, or 20 pounds gage, and the back pressure seldom less than 4 pounds absolute, so the range of pressure would be 31 pounds, or about one-sixth of the absolute boiler pressure. When this is increased by 50 percent, to allow for the fact that it is an alternating load,

we have one-quarter of the absolute boiler pressure as the equivalent load acting upon the low-pressure cylinder walls.

These assumptions are for the engine working under ordinary conditions. There are, however, other conditions which may exist at times, and for which allowance should be made. The high-pressure cylinder is liable to have water carried into it with the steam from the boiler; the medium-pressure and low-pressure cylinders may have their power increased by admitting live steam into the receivers, and so increase the pressure at which these cylinders take steam. Allowance for all these things is generally made by adding a fixed amount to the pressure assumed, so that the formula for barrels and liners is as follows:

$$t = \frac{(P + 25)D}{6{,}000} + \frac{40}{100 + D}, \qquad (15)$$

where t = thickness of walls in inches,

P = maximum pressure in cylinder, assumed as above,

and D = diameter of cylinder, as calculated.

Equation (15) may be used for cylinders without liners. When liners are used they may be calculated by (18), and the barrels made of the same thickness. When used for the liner, the term

$$\frac{40}{100 + D}$$

adds something to the thickness, to allow for reboring later, and to insure that the walls shall be of a thickness practical for casting, if the diameter D is small. When (15) is used for the barrel of a cylinder with a liner, the

$$\frac{40}{100 + D}$$

is not needed for reboring, but is required because the barrel is larger in diameter than the liner, and may be thinner in places than designed, due to the displacement of the core

during casting. Since the liner is finished inside, and at certain places outside, it will be of the designed thickness.

If jacket steam is to be used, and the pressure of the steam is to be greater than the $(P + 25)$ in the formula, the greater pressure should be used in obtaining the necessary thickness of barrel and liner.

All of the cylinder barrel thicknesses and liner thicknesses are usually made the same. The thickness for each cylinder can be found, and then an average taken, or only the thickness for the high-pressure cylinder need be figured, and all the others made the same. The thicknesses of the other parts of the cylinder and valve chest casting can be made some fraction of the liner or barrel thickness t, thus:

Thickness of cylinder bottom, single,	$= t$,
Depth of ribs of cylinder bottom, single,	$= 5t$,
Thickness of ribs of cylinder bottom, single,	$= t - {}^1/_{16}$ inch,
Thickness of cylinder bottom, double,	$= 0.9t$,
Distance between walls, double,	$= 5t$ (at least),
Thickness of cylinder flange,	$= 1.3t$ to $1.4t$,
Width of cylinder flange,	$= 2.75t$ to $3.25t$,
Thickness of metal in cylinder feet,	$= t$,
Thickness of flange on cylinder feet,	$= 1.5t$ to $1.75t$,
Diameter of bolts for cylinder feet,	$= 1.4t$ to $1.6t$,
Thickness of metal in cylinder cover, single,	$= t$,
Thickness of metal in cylinder cover, double,	$= 0.85t$,
Spacing of webs in cover and bottom	$= p = \frac{t \times 100}{\sqrt{P}}$,
Thickness of metal in valve liners,	$= t$,
Thickness of metal in ports and passages	$= 0.85t$ to $0.9t$.

The general shape of the cylinder bottom and covers will be determined by the shape of the pistons used. The length of the cylinder must be such that a piston of the desired shape

can travel the amount of the stroke, and have clearance at the top and bottom of the cylinder.

The clearances are usually as follows:

TABLE VI.

Diameter of Cylinder.	Bottom Clearance.	Top Clearance.
16 to 24 inches	1/4 to 3/8 inch	3/16 to 1/4 inch
24 to 40 inches	5/16 to 1/2 inch	1/4 to 3/8 inch
40 to 60 inches	1/2 to 9/16 inch	3/8 to 1/2 inch
60 to 80 inches	9/16 to 11/16 inch	1/2 to 9/16 inch
80 to 100 inches	11/16 to 13/16 inch	9/16 to 5/8 inch
Above 100 inches	13/16 to 7/8 inch	5/8 to 11/16 inch

PISTONS.

The coned cast steel piston is the one most used, but sometimes the cast-iron box piston is used. The latter has the shape shown in Fig. 6, and the usual proportions are as follows:

Depth of piston = $1.5d$ to $1.6d$, where d = diameter of piston rod.

Thickness of face of piston = $t = 0.0025 D\sqrt{P} + 0.33$ inch, where D = diameter of cylinder; $P = 0.5$ boiler pressure in high-pressure cylinder; $P = 0.25$ boiler pressure in medium-pressure cylinder; $P = 0.17$ boiler pressure in low-pressure cylinder.

Thickness near rim $= a = 0.9t$,

Thickness of ribs $= b = 0.9t$,

Thickness of boss around rod $= c = 1.75t$,

Thickness of junk ring flange $= e = 1.2t$,

Thickness of packing ring $= f = 0.75t$,

Breadth of packing ring $= g = 3t$,

Diameter of junk ring bolts $= \frac{t}{2} + 0.25$ inch,

Pitch of junk ring bolts $= 10$ diameters.

When cast steel pistons are used they are all made, when possible, of the same over-all depth, as shown in Fig. 7, the

details of the rim being the same for all. The slope on the under side of the low-pressure piston varies from 1 in 3 to 1 in 6; the steeper this slope is made the longer the piston rod

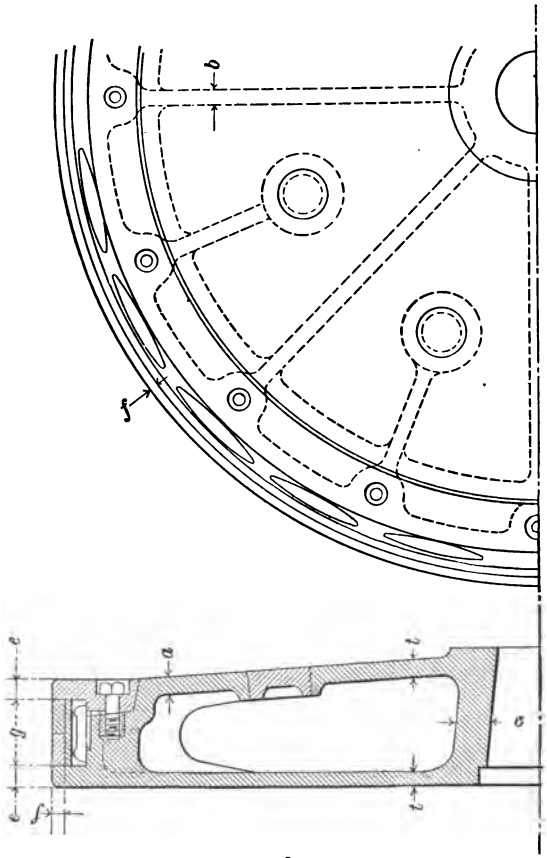


FIG. 6.

must be, and the greater the total height of the engine. The flange on the foot of the cylinder attaching it to the housings, or frame, must be sufficiently far below the under surface of

the cylinder bottom to allow the nuts on the bolts in the flanges to be set up.

Figs. 8, 9 and 10 show some of the ways in which the cylinder bottom may be constructed to fit the pistons. The construction shown in Fig. 9 can be used when the low-pressure piston has a steep slope on the underside, but the construction shown in Fig. 8 is much stronger.

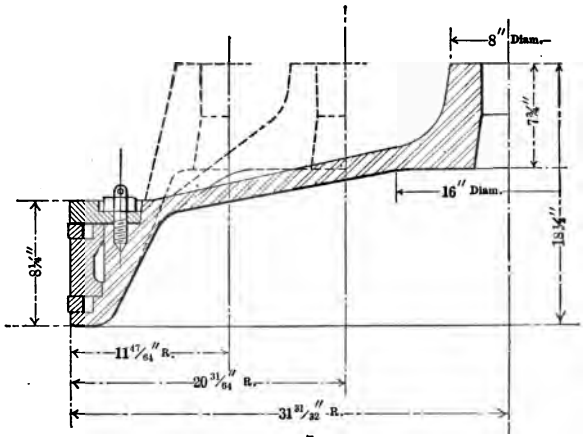


FIG. 7.

When cast steel pistons are used, the thickness at the rim and at the center line of the boss can be obtained from the following formulæ:

At the center, $$t = \frac{D}{200}\sqrt{P} + 0.25 \text{ inch.} \qquad (16)$$

At the rim, $$t' = 0.5t, \qquad (17)$$

where D = the diameter of the cylinder, and P may be taken as follows:

Ratio of Boiler Pressure for	Triple.	Quadruple.
For the high-pressure cylinder,	$P = 0.5$	$P = 0.45$
For the first medium-pressure cyl.,	$P = 0.25$	$P = 0.20$
For the second medium-pressure cyl.,		$P = 0.175$
For the low-pressure cylinder,	$P = 0.20$	$P = 0.10$

The construction of the pistons at the rim and at the boss is usually the same for all cylinders of the engine, except that, when the high-pressure cylinder is of small diameter, it is sometimes necessary to make a modification. Some of the

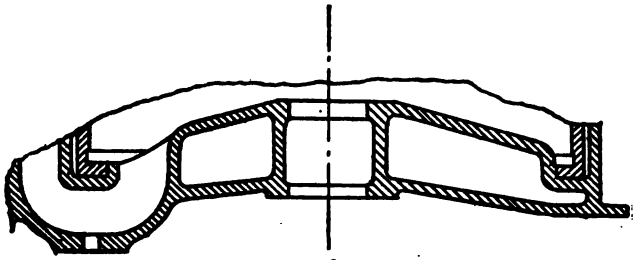


FIG. 8.

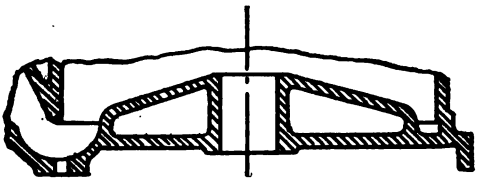


FIG. 9.

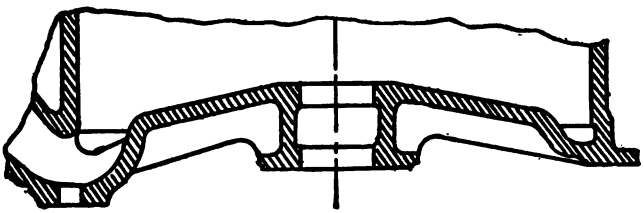


FIG. 10.

constructions used for rims are shown in Figs. 11, 12, 13 and 14. The dimensions to be used in any case should be determined by the diameter of the low-pressure cylinder. The depth of the pistons at the boss should be from 1½ D to 1¼ D; 1½ D for piston rods up to 6 inches in diameter, and

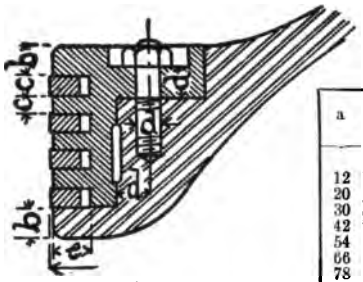


FIG. 11.

a	b	c	d	e	No. of Rings.
12	1¼	¾	1	11/16	2 or 3
20	1½	¾	1	13/16	2 or 3
30	1¾	⅞	1⅛	1⅛	3
42	1¾	⅞	1⅛	1⅜	3 or 4
54	1⅞	1	1⅛	1⅝	3 or 4
66	2	1	1¼	1 11/16	3 or 4
78	2	1 1/16	1¼	1¾	3 or 4

Dimensions in inches.

ranging from 1½ D to 1¼ D for rods from 6 inches to 11 inches in diameter. D is the diameter of the piston rod. The diameter of the boss should be sufficient to afford bearing for the nut on the end of the piston rod, the design of which will be taken up later. The underside of the piston at the center is usually made horizontal for a distance from the center equal to

$$\frac{D}{10} + 1.5 \text{ inches},$$

where D = the diameter of the low-pressure cylinder.

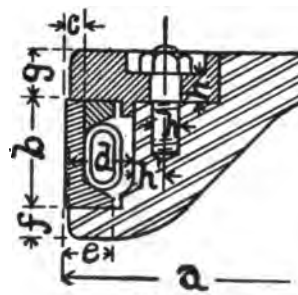
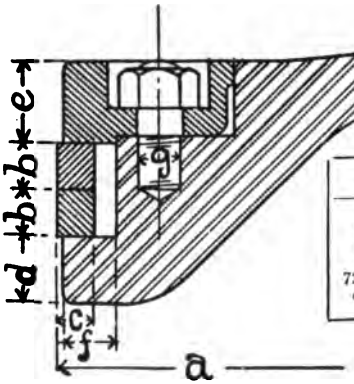


FIG. 12.

a	b	c	d	e	f	g	h
20	1⅞	5/8	1 1/2"	15/16	1¼	2¼	1
24	2⅛	11/16	1 11/16	1 1/16	1¼	2¼	1
30	2⅜	11/16	1⅞	1 3/16	1½	2½	1⅛
40	2¾	¾	2⅛	1⅜	1⅝	2½	1⅛
48	3	¾	2⅜	1½	1¾	2½	1⅛
60	3⅜	¾	2⅝	1 11/16	1⅞	2¾	1¼
75	3⅞	⅞	2⅞	1 15/16	2	2¾	1¼
100	4	⅞	3¼	2	2¼	2¾	1¼

Dimensions in inches.



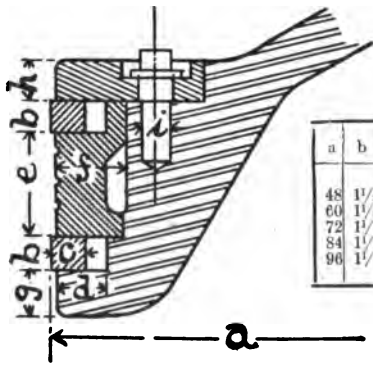
a	b	c	d	e	f	g
36	1	3/4	2	2 1/2	1 1/2	1 1/8
48	1 1/8	7/8	2 1/8	2 1/2	1 5/8	1 1/8
60	1 1/8	7/8	2 1/8	3	1 5/8	1 1/4
72 and over	1 1/4	1	2 1/4	3	1 3/4	1 1/4

Dimensions in inches.

FIG. 13.

CYLINDER COVERS.

The shape of the underside of the cylinder cover is determined by the shape of the top of the piston. The distance *a*, Figs. 15 and 16, from the underside of the cover to the bottom of the flange, is determined by the height of the steam port entering the cylinder, and the thickness of metal above the port needed for the studs of the cylinder cover. The height



a	b	c	d	e	f	g	h	i
48	1 1/8	1	1 3/4	3	2 1/2	1 1/2	1 1/2	1 1/8
60	1 1/8	1	1 3/4	3	2 1/2	1 1/2	1 1/2	1 1/8
72	1 1/4	1 1/8	1 7/8	3 1/4	2 3/4	1 5/8	1 5/8	1 1/4
84	1 1/4	1 1/8	1 7/8	3 1/4	2 3/4	1 5/8	1 5/8	1 1/4
96	1 1/4	1 1/8	2	3 1/2	3	1 3/4	1 3/4	1 1/4

Dimensions in inches.

FIG. 14.

of the port is generally from 4½ to 7 inches, depending upon the area necessary in order that the steam may enter with the desired speed.

The breadth of the joint between the cover and the barrel should be from 2.75 d to 3.25 d, being usually 3 d, where d is the diameter of the stud connecting the flanges. The studs should be spaced far enough apart to allow a wrench to be used freely, and near enough together to make the joints steam

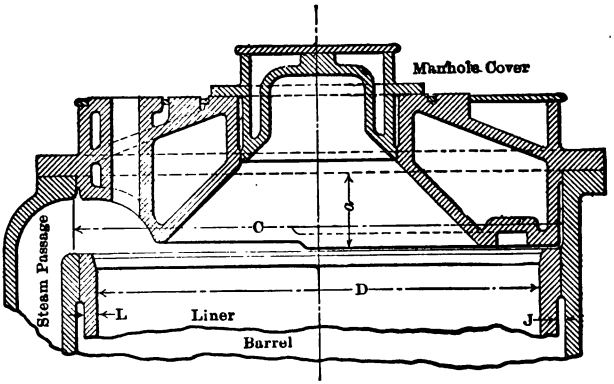


FIG. 15.

tight. For the sake of convenience and simplicity, it is usual to calculate the studs which should be used for the high-pressure cover, and then to use the same size of stud upon the other cylinder covers, increasing the spacing as the steam pressure decreases.

When a liner is used the inner surface of the barrel will have a diameter equal approximately to

$$C = D + 2L + 2J, \qquad (18)$$

where D = the diameter of the cylinder,

L = the thickness of the liner,

J = the width of the jacket space, usually ¾ inch or 1 inch.

The load upon the cover will be the area of a circle of diameter C multiplied by the gage boiler pressure:

$$\text{Load on cover, } = \frac{\pi C^2}{4} \times (B.P.) \qquad (19)$$

The cover studs should be spaced from 2.75 d to 3.25 d on the high-pressure cylinder, and the diameter of the pitch circle for these studs, upon the assumption that the width of joint

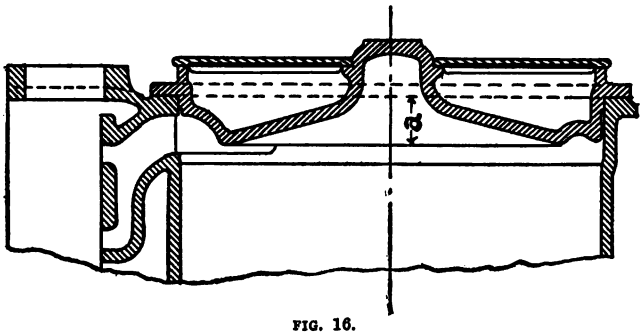


FIG. 16.

is to be 3 d, will be $C + 3\ d$. The total load to be carried by the studs being known, a diameter of stud d, equal approximately to the thickness of the barrel, should be selected. Table IV. gives the load that the stud can carry, so that the total number necessary can be found.

$$\frac{(C + 3d)\pi}{\text{number of studs}} = \text{spacing of studs,}$$

which should be from 2.75 d to 3.25 d. If the studs cannot be brought near enough together except by taking d less than ¾ inch, the diameter is taken as ¾ inch, and enough studs used to bring the spacing right. The spacing upon the medium-pressure cylinder cover should be from 4 d to 5 d, and upon the low-pressure cover from 5 d to 6 d. It is well to tabulate the results, as shown later in the calculations.

The nuts used on the cover studs are often thicker than the standard nut, in order that they may stand frequent handling. Covers for cylinders over 35 inches in diameter should have manholes in them, usually in the center of the cover. These manholes should be not less than 16 inches in diameter, and constructed as shown in Fig. 15. Manholes are frequently placed in the bottom of the cylinders, to give access to the underside of the piston without taking off the cover and removing the piston. These have to be placed where they will clear the upper end of the connecting rod.

CALCULATIONS FOR CYLINDERS AND PISTONS.

The use of the formulæ given will be illustrated by continuing the calculations for the engine whose cylinder diameters and cut-offs have been previously determined.

The assumptions and calculations made so far give us

I. H. P. = 3,000, P. S. = 850 feet per minute, B. P. = 185 pounds gage

Cylinder diameters and stroke, $23\frac{1}{2}''$, $41''$, $64'' \times 42''$.

Cut-offs: H. P. = 0.675, M. P. = 0.60, L. P. = 0.65.

Cylinders.—Formula (15):

H. P. cylinder liner and barrel, $$t = \frac{(185 + 25)23.5''}{6{,}000} + \frac{40''}{123.5}$$
$$= 1.147''(\text{Use } 1\tfrac{1}{8}'')$$

M. P. cylinder liner and barrel, $$t = \frac{(100 + 25)41''}{6{,}000} + \frac{40''}{141}$$
$$= 1.136''(\text{Use } 1\tfrac{1}{8}'')$$

L. P. cylinder liner and barrel, $$t = \frac{(50 + 25)64''}{6{,}000} + \frac{40''}{164}$$
$$= 1.044''(\text{Use } 1\tfrac{1}{16}'')$$

Make the thickness of all cylinder barrels and liners $1\frac{1}{8}''$.

Use double walls in covers and bottoms.

Thickness of cylinder bottom

$$= 0.9 \times 1.125'' = 1.01'' \text{ (Use } 1''\text{).}$$

Distance between walls of cylinder bottom

$$= 5 \times 1.125'' = 5.625'', \text{ at least.}$$

Thickness of cylinder flanges

$$= 1.3 \times 1.125'' = 1.46'' \text{ (Use } 1\tfrac{1}{2}''\text{).}$$

Width of cylinder flanges $= 3\ d$.

Thickness of metal in cylinder feet

$$= 1.125''.$$

Thickness of flanges on cylinder feet

$$= 1.5 \times 1.125'' = 1.69'' \text{ (Use } 1\tfrac{3}{4}''\text{).}$$

Diameter of bolts for cylinder feet

$$= 1.4 \times 1.125'' = 1.58'' \text{ (Use } 1\tfrac{5}{8}''\text{).}$$

Thickness of metal in cylinder cover

$$= 0.85 \times 1.125'' = 0.955'' \text{(Use } {}^{15}/_{16}''\text{).}$$

Maximum spacing of webs in bottom and cover:

$$\text{H. P.,} \quad \frac{1.125'' \times 100}{\sqrt{185}} = 8.3''. \quad \text{(Use } 8\tfrac{1}{2}''.\text{)}$$

$$\text{M. P.,} \quad \frac{1.125'' \times 100}{\sqrt{100}} = 11.25''.$$

$$\text{L. P.,} \quad \frac{1.125'' \times 100}{\sqrt{50}} = 15.9''. \quad \text{(Use } 16''.\text{)}$$

Thickness of metal in valve liners

$$= 1.125''.$$

Thickness of metal in ports and passages

$$= 0.9 \times 1.125'' = 1.01''. \quad \text{(Use } 1''.\text{)}$$

Piston clearances:	H. P.	M. P.	L. P.
Top.....	$\frac{3}{8}''$	$\frac{3}{8}''$	${}^{9}/_{16}''$
Bottom.....	$\frac{1}{2}''$	$\frac{1}{2}''$	${}^{11}/_{16}''$

Pistons.—Formula (16):

$$\frac{23.5''}{200} \times \sqrt{100} + 0.25'' = 1.43'', \text{ high-pressure.}$$

$$\frac{41''}{200} \times \sqrt{50} + 0.25'' = 1.70'', \text{ medium-pressure.}$$

$$\frac{64''}{200} \times \sqrt{40} + 0.25'' = 2.03'', \text{ low-pressure.}$$

Formula (17): $0.5 \times 1.43'' = 0.715''$, high-pressure.
$0.5 \times 1.70'' = 0.85''$, medium-pressure.
$0.5 \times 2.03'' = 1.015''$, low-pressure.

Depth of piston at boss $= 1.5 \times 5.25'' = 7.87''$. (Use $7\frac{7}{8}''$.)

Diameter of cylindrical nut on top of piston rod $= 4'' \times 1.75 = 7''$ (See below.) Top of boss to be 8 inches diameter.

Radius of horizontal portion of the underside of piston

$$= \frac{64}{10} + 1.5'' = 7.9''. \text{ (Use } 8''.)$$

The diameter of the body of the pistons will be:

High-pressure,	$23\frac{1}{2}'' - {}^{1}/_{32}''$	$= 23^{15}/_{32}''$.
Medium-pressure,	$41'' - {}^{1}/_{32}''$	$= 40^{31}/_{32}''$.
Low-pressure,	$64'' - {}^{1}/_{16}''$	$= 63^{15}/_{16}''$.

The details of the piston are shown in Figure 7.

Bolting for Cylinder Covers.—Formula (18):

$$C = 23.5'' + 2 \times 1.125'' + 2 \times 0.75'' = 27.25''.$$

$$\text{Load on cover} = \frac{\pi(27.25)^2}{4} \times 185 = 108{,}000 \text{ pounds.}$$

Try $1\frac{1}{8}$-inch studs of 70,000 pounds ultimate strength, which (see Table IV.) can carry 3,115 pounds each.

$$\frac{108{,}000}{3{,}115} = 35. \quad \text{(Approximately.)}$$

$$27.25'' + 3 \times 1.125'' = 30.625'';$$

Circumference of pitch circle $= 30.625 \times \pi = 96''$; $\frac{96''}{35} = 2.75''$;

$\frac{2.75''}{1.125''} = 2.44$ diameters; (Too close).

Try 1¼-inch studs. Working load = 4,110 pounds.

$$\frac{108{,}000}{4{,}110} = 26. \quad \text{(Approximately.)}$$

$$27.25'' + 3 \times 1.25'' = 31''; \quad 31'' \times \pi = 97.3'';$$

$\frac{97.3''}{26} = 3.75''$; $\frac{3.75''}{1.25''} = 3$ diameters; (Use 1¼-inch studs).

Cylinder Cover Studs. Cylinder.	Diameter.	Diameter of Stud.	Pitch Circle of Studs.	Number of Studs.	Spacing of Studs, Diameters.
High.....	23.5″	1.25″	31″	26	3.
Medium...	41″	1.25″	48½″	29	4.2
Low.....	64″	1.25″	71½″	33	5.44

RECIPROCATING MEMBERS.

The reciprocating parts of a marine engine are usually designed for the maximum load to which the parts will be subjected. This load cannot be determined before the engine is built, as it depends largely upon the valve setting, and the indicator card is needed in order that the steam load may be determined. The maximum load can be determined approximately, however, by means of factors obtained from the analysis of the results of engine trials. If *I. H. P.* is the indicated horsepower developed in a cylinder with a piston

speed per minute $= P.\ S.$, then the mean total load acting upon the piston

$$= \frac{I.H.P. \times 33{,}000}{P.S.}.$$

It has been found in the results analyzed that the maximum load was about 1.93 of this mean theoretical load. The formula to be used here for finding the maximum load for which the reciprocating parts are to be designed is, therefore,

$$W = \frac{2 \times I.H.P. \times 33{,}000}{P.S.}, \qquad (20)$$

where $I.\ H.\ P.$ is the indicated horsepower to be developed in the cylinder in question. It will be found that this gives very nearly the same results as the formula often used, $W =$ high-pressure area $\times$ boiler pressure.

Piston Rods.—Piston rods are usually of the shape shown in Figs. 17, 18, 19 and 20, with a parallel middle body subjected to alternating loads, shouldered at each end of the parallel body to take the thrust, and threaded portions beyond to take the tension, the latter portions being subjected to intermittent loads. A column formula must be used for the design of the middle portion, and in these calculations a formula based upon Rankine's long column formula will be used.

The formulæ for solid and hollow rods are as follows:

Solid rods, $$D^2 = \sqrt{\frac{0.48FCl^2}{10{,}000{,}000} + F^2} + F; \qquad (21)$$

Hollow rods, $$D^2 = \sqrt{\frac{0.48FCl^2}{10{,}000{,}000} + F^2 + (2F + d^2)d^2} + F; \qquad (22)$$

where $D =$ the diameter of the rod in inches at the middle of its length;

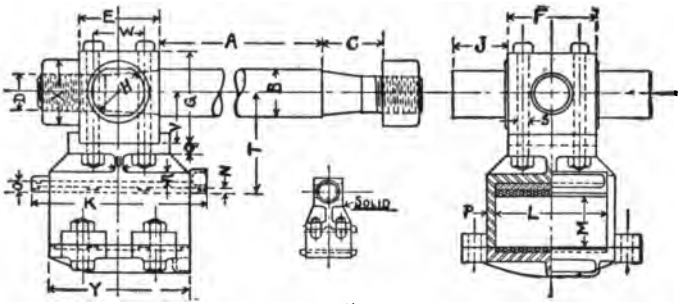


FIG. 17.

PISTON ROD, CROSSHEAD AND BOX GUIDE.

A	B	C	D	E	F	G	H	J	K	L	M
3′ 2″	3 1/2″	.....	3″	7″	7 1/2″	7″	4 1/2″	5 1/2″	14″	9″	.....
3′ 7″	3 3/4″	5 1/2″	3″	7″	8″	8″	4 1/2″	4 3/4″	15 1/2″	10″	4 1/2″
4′ 0 5/8″	3 7/8″	5 3/4″	3 1/4″	7″	7″	8″	4 1/2″	4 5/8″	17″	8″	4 1/2″
3′ 11 1/4″	4 1/4″	7 1/2″	3 1/2″	8″	8″	8 1/2″	4 3/4″	5 3/8″	16″	11″	5″
.....	5″	.....	3 3/4″	9 1/2″	9 1/2″	9 1/2″	6″	6 5/8″	21″	12 1/8″	6 1/2″
4′ 10″	6 1/2″	7 1/2″	5 1/2″	10″	10″	10″	7 3/4″	7 1/2″	22″	16″	6 1/2″
4′ 6″	4 7/8″	7 1/2″	3 3/4″	8 1/2″	8″	8″	5 1/4″	6 3/4″	17 1/2″	13″	4 3/4″
2′ 10″	2 5/8″	4″	2″	5″	5″	5″	3 1/2″	4″	10″	7″	3″

N	O	P	Q	R	S	T	U	V	W	X	Y
7/16″	1 1/4″	.....	.....	7/8″	1 1/8″	.....	.....	.....	.....	.....	.....
3/8″	7/8″	5/8″	7/8″	3/4″	1″	8 1/2″	3/4″	4 3/8″	4 1/2″	5 3/4″	12 1/2″
3/8″	1″	7/8″	1″	7/8″	1″	4 1/4″	7/8″	4 1/2″	4 1/4″	5 3/4″	14″
3/8″	1 1/8″	7/8″	1″	15/16″	1 1/8″	10″	7/8″	4 1/2″	5″	6 1/8″	13″
1/2″	1 1/2″	1 1/4″	1 1/2″	.....	1 1/4″	14 1/2″	7/8″	4 3/4″	5 1/2″	.....	21″
7/8″	1 1/2″	1″	Solid	1″	.....	14 1/4″	1 1/4″	.....	.....	.....	22″
1/2″	.....	.....	Solid	.....	1 1/8″	12 1/2″	.....	.....	.....	6 3/4″	13″
.....	3/4″	3/4″	Solid	1″	3/4″	7″	1 1/4″	.....	.....	.....	8″

$$F = \frac{2W}{\pi f};$$

W = load on rod (Formula 20);

f = allowable stress per square inch $= \frac{C}{N}$;

C = the ultimate strength of the material,

= 60,000 to 70,000 pounds for merchant work,

= 80,000 to 100,000 pounds for naval work,

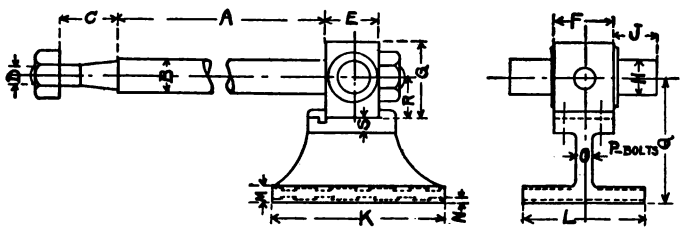


FIG. 18.

PISTON ROD, CROSSHEAD AND SINGLE SLIPPER GUIDE.

A	B	C	D	E	F	G	H	J
4′ 7¼″	5″	.....	4½″	9″	9½″	9″	6″	7″
5′ 6½″	5¾″	.....	4¾″	10¼″	11″	11″	7¼″	5½″
5′ 0½″	5¾″	.....	5″	11″	10⅛″	10½″	7¼″	8½″
5′ 5⅝″	7″	9″	5½″	12″	13″	12″	7½″	9″
7′ 7″	7½″	.....	5¼″	13½″	12⅜″	13½″	9½″	10″
5′ 8¹³⁄₁₆″	7½″	.....	6½″	14″	14″	14″	10½″	11¼″
9′ 2¾″	8½″	.....	5″	16″	17″	18½″	11″	13½″
5′ 3¹⁵⁄₁₆″	8⅛″	10″	6½″	14½″	15″	14″	9″	11″
4′ 0⅞″	4½″	4½″	3¼″	7″	7½″	8¾″	5″	5⅝″

K	L	M	N	O	P	Q	R	S
21″	12½″	2½″	7⁄16″	3¼″	4′ 1⅛″	.....	.....	.....
27″	13½″	2¾″	½″	3″	4′ 1⅜″	.....	.....	.....
24″	16″	3″	½″	3″	4′ 1⅜″	.....	.....	.....
23½″	19″	2½″	½″	2¾″	6′ 1¼″	17½″	6½″	1½″
30″	22½″	3¼″	½″	3″	4′ 1¾″	.....	.....	.....
30″	26″	3½″	½″	3½″	4′ 2″	.....	.....	.....
44″	25½″	3¾″	½″	4″	4′ 2¼″	26″	10″	2½″
26″	24″	3″	½″	3¼″	4′ 2″	19½″	7″	1⅝″
16″	14″	1¾″	7⁄16″	1¼″	2′ 1½″	12¾″	5″	1⅛″

N = factor of safety = 8 to 10 for torpedo boat engines,
= 12 to 14 for other naval engines,
= 15 for merchant ship engines;

l = length of rod in inches from under side of piston to upper side of crosshead block = (approximately) stroke + diameter of high-pressure cylinder + X, where X = 2 to 8 inches for merchant ship engines, = − 2 to − 8 inches for naval engines;

d = inside diameter of hollow rod, and may be taken from $\frac{1}{2}\sqrt{2F}$ to $\frac{2}{3}\sqrt{2F}$.

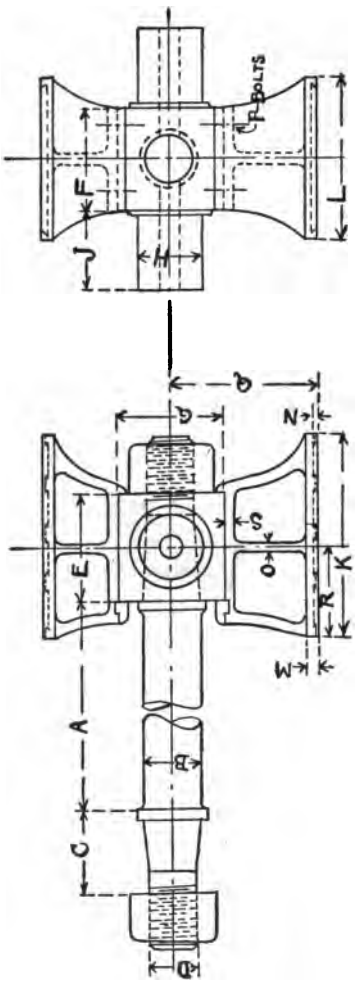


FIG. 19.

PISTON ROD, CROSSHEAD AND DOUBLE SLIPPER GUIDE.

A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S
5′ 2″	6¼″	9″	5¼″	11½″	10¾″	11½″	7″	8½″	22″	17″	1½″	½″	1″	4′	1¼″	14″	10″	1¼″
5′	4″	.....	.....	9¾″	10″	9½″	5″	6½″	19″	12″	1¼″	½″	⅞	.....	.....	12″	9½″	1¼″
5′ 5¾″	6½″	.....	5¼″	10″	11″	10″	6½″	7¾″	24″	12″	1⅝″	½″	1″	4′	1⅛″	.....	.....	.....

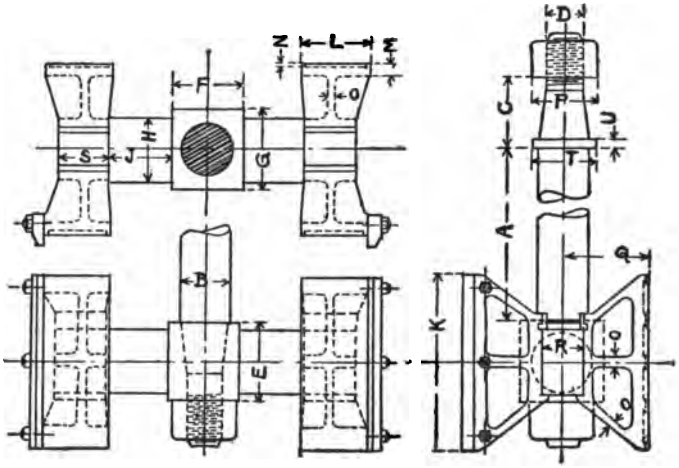


FIG. 20.

PISTON ROD, CROSSHEAD AND FOUR SLIPPER GUIDE.

A	B	C	D	E	F	G	H	J	K
8′ 8¼″ 9′ 6″	8″ 11″	11½″ 13″	6″ 8¾″	13″ 16½″	11½″ 15¾″	13″ 16¾″	10½″ 13⅜″	10″ 16½″	28″ 47″

L	M	N	O	P	Q	R	S	T	U
11″ 20″	2″ 2¼″	.... ¾″	1″ 1″	10½″ 13¼″	14″ 17¾″	7″ 8½″	8″ 14″	10″ 12″	1½″ 1¼″

In discussing the factors of safety to be used for different kinds of loads, it was stated that 12 would be used for alternating loads. It was also stated that these factors would be modified sometimes by reason of the conditions under which the part worked. The piston rod is guided more or less rigidly at three points: at the piston by the piston rings, at the stuffing-box, and at the cross-head. These three points may be out of line in the first place, or may get out of line as the engine works, and thus cause the rod to have an initial flexure at times. The factor 0.48 in the formula assumes that the rod is held rigidly at the cross-head end, and that only

two-thirds of its length is subjected to bending, whereas it is probable that it is not held rigidly at the cross-head. A third reason for modifying the factor is that the rod is working a part of the time in steam and part in the open air, and runs through a stuffing-box, which may get hot and grip it. When all of these conditions are considered it seems wise to increase the factor of safety from 12 to 15 for this member.

The thrust of the rod is transmitted to the piston and cross-head block by means of a shoulder, either positive, as in Figs. 19 and 20, or negative, as in Figs. 17 and 18. This shoulder, when negative, has a breadth of 1/16 inch in the smallest rods and 3/16 in the largest rods. When the shoulder is positive it is generally broader, varying from ¼ inch to ½ inch. The threaded portion of the rod, having to transmit tension only, is designed to have sufficient area at the root of the threads to transmit the load with a factor of safety of 10. As four threads per inch are generally used, the diameter of this portion must be increased by

$$\frac{1.299}{4} = 0.325 \text{ inch (about)};$$

to allow for cutting the threads. The diameter of the threaded portion, taken to the nearest ¼ inch, should be:

$$D' = \sqrt{\frac{40W}{\pi C}} + 0.325. \qquad (23)$$

(W and C are as above.)

The connection between the shoulder and the threaded portion is usually made by means of a tapered part, the taper being 2 or 3 inches per foot. Cylindrical nuts are generally used in attaching the piston and the cross-head.

The length of the rod between shoulders will be given only approximately by the formula L = stroke + high-pressure diameter + X. This is near enough for calculation, but the final length should be determined from the assembly drawing.

Bearing Pressures.—Before proceeding further with the design of the reciprocating parts, it will be well to take up the question of pressures allowed upon bearing surfaces. It is usual to figure the bearing surface of pins and shafts as equal to the length of the surface multiplied by the diameter of the pin or shaft, *i. e.,* as equal to the *projected* area of the bearing. The pressure allowed must be low enough to keep the bearing cool, and varies with the conditions and the possibility of artificial cooling. When there is no motion between two parts the pressure between them can be very large, but as the extent of the motion between them increases the pressure allowed decreases. Since it is a question of keeping the bearing cool, we can deal with the mean unit bearing pressure, since the maximum will occur generally but for an instant. In some cases, however, it is more convenient to deal with the maximum pressure to avoid confusion. The pressures allowed upon the different surfaces are shown in Table VII. The most of the pressures are based upon the maximum loads, as these loads are used in the design of the other parts, and it

TABLE VII.

Allowable pressure on bearing surfaces in pounds per square inch of projected surface:

	Merchant.	Naval.
USING MEAN LOADS—		
Crank Pin	200 to 250	250 to 300
Main Bearings	200 to 350	250 to 500
USING MAXIMUM LOADS—		
Slipper Guide	60 to 80	70 to 100
Crosshead Pins	850 to 1,200	1,200 to 1,800
Link Block Pin	750 to 1,000	850 to 1,200
Link Block Gibs	250 to 400	350 to 500
Eccentric Rod Pins	700 to 950	900 to 1,100
Drag Rod Pins	500 to 700	700 to 800
Eccentrics	150 to 200	175 to 225
Thrust Collars	50 to 80	80 to 100

is less confusing to deal with but one load. It will be noticed that those surfaces which have but little relative motion have the greatest unit pressure allowed upon them, while those with more motion, such as the slipper, main bearings and crank pins, have a lower unit pressure allowed. The question of the determination of the loads coming upon the different surfaces will be taken up later

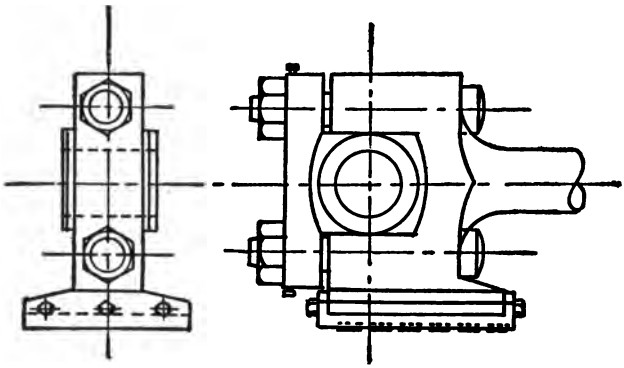


FIG. 21.

Cross-Head Block.—There are two types of cross-heads, those with external bearing surface, as shown in Figs. 17, 18, 19 and 20, and those with internal bearing surface, as shown in Fig. 21. When the bearing surface is external, the block is approximately a cube, with the two cross-head pins projecting from two opposite faces. When the bearing surface is internal, the cross-head consists of a box, with bottom brass and cap, in which works a pin held fast in the jaws of the connecting rod fork. (See Fig. 22.) This latter type is used principally upon small, fast running engines, and upon torpedo boat engines.

The cross-head block with external bearing surface is generally from 1.8 D to 1.9 D upon a side (D being the diameter of the piston rod), and tends to split upon a plane

passing through the axis of the piston rod, where the cross-section of metal resisting the bending moment is shaped as shown in Fig. 23. The bending moment upon this section will be

$$M = \frac{WL}{4},$$

where W is the load previously found, and L is the distance from center to center of cross-head pins.

Cross-Head Pins.—The maximum load W, divided by P, the allowable pressure upon the cross-head pins, taken from

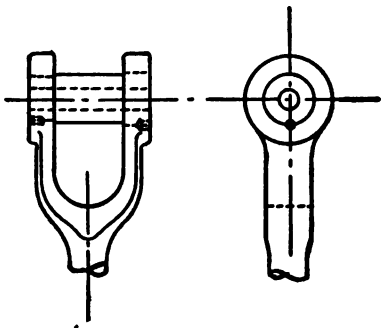


FIG. 22

Table VII., will give the surface that must be provided in these pins, or

$$l \times d = \frac{W}{2P},$$

where l = the length of one pin and d = its diameter. The diameter d should be from 1.2 D to 1.4 D, where D is the diameter of the piston rod. This will give a pin with a sufficiently strong attachment to the block, so that no other calculation need be made upon it. This diameter of pin may have to be changed later on, when the dimensions of the upper end of the connecting rod are known, but it is sufficiently

close for calculation. The length of the pin will usually come about equal to its diameter.

The breadth of the block having been assumed, and the length of the pins found, the bending moment $\frac{WL}{4}$ can be found.

The moment of inertia of the section resisting the bending is

$$I = \frac{bh^3}{12},$$

where b is the net breadth after taking out the piston rod hole, and h is to be found so that the stress shall be reasonable.

$$h = \sqrt{\frac{3WL}{2bf}}. \qquad (24)$$

Here f should be chosen so that the factor of safety will be 12. The cross-head blocks are usually made of wrought steel, or cast steel, with the pins hardened. A facing strip should be allowed around each pin, from 1 inch to 2 inches wide, and projecting from the block 1/16 to ⅛ inch.

If the cross-head has internal bearing surface, the length and diameter of the bearing are calculated in the same way as are the pins,

$$l \times d = \frac{W}{L}.$$

The length of the pin determines the thickness of the block, and the other details are figured in the same way as are the boxes of the connecting rod, whose design will be taken up later.

Slipper.—The slippers attached to the cross-heads are usually one of the four types shown in Figs. 17, 18, 19 and 20. Fig. 17 shows the type used upon many small engines of 1,000 indicated horsepower and less. It is a cast-steel box,

inclosing for a guide a square cast-iron beam, upon whose front and back surface the slipper bears. There are some parts of the slipper which should be figured, such as the amount of surface and sizes of bolts, but there are other parts which are determined by practical considerations, and dimensions such as *P, R, O, Q, U,* etc. (see table accompanying Fig. 17), can be taken from blue prints of other engines, or from the figures here given.

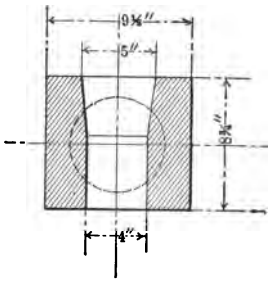


FIG. 23

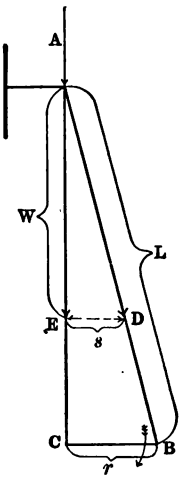


FIG. 24.

The dimensions *K* and *L* of the bearing surface are determined from the total load coming upon the guide and the allowable unit pressure given by Table VII. The maximum pressure comes upon the slipper when the crank is 90 degrees from the line of dead points. Referring to Fig. 24, it will be seen that when a load $W = A\ E$ acts through the piston rod *A A*, and is transmitted to the connecting rod *A B*, a load *E D* will come upon the slipper. This load

$$S = W \times \frac{ED}{AE},$$

or, approximately,

$$G = W \times \frac{ED}{AD} = W \frac{r}{l}$$

where $\frac{r}{l}$ is the ratio of the crank length to the length of the connecting rod. This ratio usually is from ¼ to 1/5 for marine engines. The amount of surface for the slipper will then be

$$K \times L = \frac{W \times r}{l \times P}. \qquad (25)$$

The tables under Figs. 17, 18, 19 and 20 will show that K and L are nearly equal in smaller engines, but that $K = 1.5\ L$ in the larger engines. The bolts joining the slipper to the cross-head, and those joining the two parts of the box, must be figured for the load G. The distance from the center line of the piston rod to the surface of the slipper, see Q, Figs. 17 to 20, must be sufficient to have the boxes on the forked end of the connecting rod clear any adjacent parts or surfaces when angled over the extreme amount. The exact distance must be taken from the assembly drawing, but it will generally be from 2.5 D to 3 D.

The cast-iron beam upon which the box guide slides can be figured as a beam supported at the ends, and loaded at the middle with a load G. Its length will equal the stroke of the engine + K. The thickness of the beam M can be assumed from the table, and the beam figured as though of solid cross-section, although it will be cored out inside to permit water to circulate and carry off the heat.

The type of slipper shown in Fig. 18 is the one most commonly used. The amount of surface needed when going ahead is figured in the same way as in the previous case, but

as the "backing" surface is upon the back of the slipper, it will be less than the "go-ahead" surface, because of the web joining the slipper to the block flange. The "backing" surface is made as great as possible, but is usually only about two-thirds of the "go-ahead" surface. When backing, there is a tendency for the slipper to break close to the web spoken of, so that it must be figured for bending at that point.

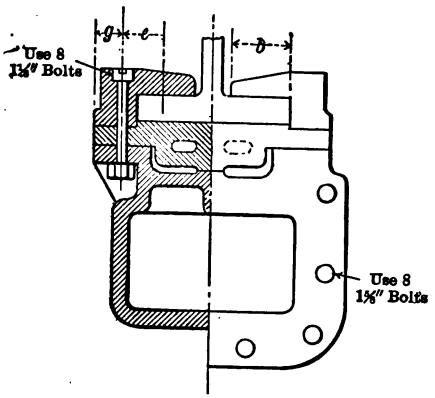


FIG. 25

The arrangement of slipper and backing guide is shown in Fig. 25. It is assumed that all of the restraining pressure exerted by one backing guide $= \frac{G}{2}$, is acting at the center of the surface, with an arm $\frac{b}{2}$. The front and back of the slippers are recessed to take white metal, so that the thickness effective for resisting bending will be that between the backs of the white metal. Choosing a stress appropriate for the material of which the slipper is to be made, generally cast or wrought steel, the necessary thickness of slipper between the backs of the white metal will be

$$t = \sqrt{\frac{W \times r \times b \times 6}{2 \times l \times 2 \times K \times f}}, \qquad (26)$$

where W = load upon piston rod,

r = length of crank,

b = breadth of backing surface in one guide,

l = length of connecting rod,

K = length of slipper,

f = a stress which will allow a factor of safety of 12, since the slipper is subjected to loads acting in opposite directions.

The thickness thus found must be increased by the thickness of white metal on each face.

The backing guide can be figured in the same way, by merely changing the allowable stress, as cast iron is usually employed for the backing guide, and it is assumed that only that portion of the guide in immediate contact with the slipper resists the bending; therefore, K has the same value as before. Since the bending takes place where there is a sharp corner, it is well to use a low stress, about 1,500 pounds for cast iron. The bolts attaching the backing guide to the engine frame are spaced from 6 to 8 diameters apart. These bolts are brought into play only when the engine is backing, and it will be assumed that all of the bolts in the guide help to carry the load. This load will be

$$\frac{G}{2} \times \frac{e + g}{g} \text{ (see Figure 25)},$$

the distance $e + g$ depending upon the size of bolt used, and this size will have to be determined by trial. The number of bolts will be

$$\frac{K + s}{n},$$

where K = length of slipper, s = length of stroke, and n = spacing of bolts. These bolts can be placed within a half inch of the edge of the slipper. Therefore,

$$e = \frac{b}{2} + \frac{1''}{2} + \frac{d}{2},$$

where d = diameter of bolt assumed. The bolt should be placed not nearer to the outer edge of the guide than 1.5 d, and the projecting lip will be from ½ inch in small engines to 1 inch in larger engines, so that $g = 1.5\ d + ½$ inch to 1 inch. The effect of placing the bolts so far to one side of the line of action of the force will be to increase the effective load $\frac{G}{2}$ to about G, and this can be kept in mind when selecting the trial size of bolt from Table IV.

When the cross-head block is provided with two slippers, as shown in Fig. 19, there is as much "backing" surface as "go-ahead" surface, and a guide must be provided for both sides of the engine frame. In this case the back of the slipper does not have to be kept open to clear the "backing" guide, so additional webs can be put in, thus making a much stronger and stiffer slipper. Since the slippers are not subjected to bending, they can be made thinner than in the previous case; and the webs, joining the slipper to the flange which connects it to the block, are made thinner than the single web. These thicknesses cannot be figured, but are determined by practical considerations; the parts must be thick enough to "look well," and have the requisite stiffness. The data given with Fig. 19 will be useful in this respect.

The cross-head shown in Fig. 20 is used on very large engines. The "backing" surface is equal to the "ahead" surface, and all of the parts are designed in the same way as those previously mentioned. This design enables the columns

to be placed closer to the center line of the engine, and is used generally with four cast-steel columns to each cylinder, upon each of which is a guide surface.

Upon all of the types of cross-heads it will be noticed that provision is made for removing the slipper without disturbing the alinement of the other parts. The key at the top of the block, which permits this removal, is held in place by two or more bolts. In the "two-slipper" and "four-slipper" type of guide, the bolts attaching the slipper to the block need be only ¾ inch to 1 inch in diameter. In the other types they should be figured to carry the full load coming upon the slipper when backing. The key at one end and the lip at the other prevent any shear from coming upon these bolts.

CALCULATIONS FOR PISTON ROD, CROSS-HEAD AND SLIPPER.

Formula (20): $W = \dfrac{2 \times 1{,}000 \times 33{,}000}{850} = 77{,}700$ pounds.

Assume $C = 70{,}000$ pounds, $l = 42 + 23.5 + 4 = 69.5''$, (Use $70''$)
$N = 15$.

Then $f = \dfrac{70{,}000}{15} = 4{,}670$ $\quad F = \dfrac{2 \times 77{,}700}{3.142 \times 4{,}670} = 10.6$

Formula (21):

$$D^2 = \sqrt{\frac{0.48 \times 10.6 \times 70{,}000 \times (70)^2}{10{,}000{,}000}} + (10.6)^2 + 10.6$$

$$= \sqrt{174.5 + 112.5} + 10.6 = 16.9 + 10.6 = 27.5 \qquad D = 5.25$$

Formula (23):

$$D' = \sqrt{\frac{40 \times 77{,}700}{3.142 \times 70{,}000}} + 0.325 = 4.085'', \text{ (Use } 4''\text{)}.$$

Allowing a shoulder of ⅛ inch at each end of the rod, and tapering at the rate of 3 inches per foot, the ends of the rod will be as shown in Fig. 26. Assume breadth of cross-head

block $= 1.8 \times 5.25$ inch $= 9.45$ inches (use $9\frac{1}{2}$ inches). Cross-head pins will be $5.25 \times 1.2 = 6.30$ inches (use $6\frac{1}{4}$ inches).

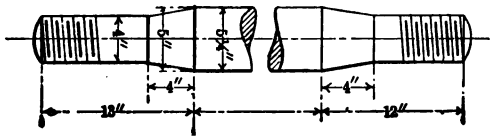


FIG. 26.

Assume allowable unit pressure on pins to be 1,000 pounds per square inch.

$$l \times d = \frac{77{,}700}{2 \times 1{,}000} = 38.85, \quad l = \frac{38.85}{6.25} = 6.22, \text{ (Use } 6\tfrac{1}{4}\text{)}.$$

$$L = 9.5'' + 6.25'' = 15.75''.$$

The crosshead will be made of wrought steel, so that f can be

$$\frac{60{,}000}{12} = 5{,}000 \text{ pounds}, \quad b = 9.5'' - \frac{4'' + 5''}{2} = 5''. \text{ (See Fig. 23.)}$$

Formula (24): $$h = \sqrt{\frac{3 \times 77{,}700 \times 15.75}{2 \times 5.0 \times 5{,}000}} = 8.57'', \text{ (Use } 8\tfrac{3}{4}\text{)}.$$

Assume $\frac{r}{l} = \frac{I}{4.5}$; $G = \frac{77{,}700}{4.5} = 17{,}250$, P will be taken as 70 pounds.

The slipper is to be like Figure 18, made of cast steel,

$$f = \frac{60{,}000}{12} = 5{,}000 \text{ pounds}.$$

Formula (25): $$K \times L = \frac{77{,}700}{4.5 \times 70} = 246 \text{ square inches}.$$

Let $L = 13''$ and $K = 19''$.

Assume that the web joining the slipper to the block flange, and

clearances, will take up 3 inches. The two backing surfaces will then be each 5 inches wide.

Unit pressure when backing $= \frac{17{,}250}{10 \times 19} = 91$ pounds.

Formula (26): $t = \sqrt{\frac{77{,}700 \times 5 \times 6}{2 \times 4.5 \times 2 \times 19 \times 5{,}000}} = 1.167''$,

(Use $1\frac{1}{4}''$). Add $\frac{1}{2}$ inch white metal for each face.

Total thickness of slipper $= 2\frac{1}{4}$ inches.

Backing guide to be of cast iron, stress $= 1{,}500$ pounds.

Formula (26):

$$t = \sqrt{\frac{77{,}700 \times 5 \times 6}{2 \times 4.5 \times 2 \times 19 \times 1{,}500}} = 2.13'', \text{ (Use } 2\tfrac{1}{8}''\text{)}.$$

Assume backing guide bolts to be $1\frac{1}{4}$ inch diameter. Each guide carries a load of $\frac{77{,}700}{4.5 \times 2} = 8{,}633$ pounds.

$$e = \frac{5''}{2} + 0.5'' + \frac{1.25''}{2} = 3.625''; \quad g = 1.25 \times 1.5 + 0.75 = 2.625'';$$

$$\frac{e + g}{g} = \frac{3.625 + 2.625}{2.625} = 2.38.$$

1 bolts are spaced $7 \times 1.25'' = 8.75''$ apart, the number needed will be $\frac{42 + 19}{8.75} = 7$ (approximately). The load upon each bolt will be $\frac{8{,}650 \times 2.38}{7} = 2{,}945$ pounds. From Table IV., a bolt $1\frac{1}{4}$ inch in diameter can carry 3,520 pounds. Try $1\frac{1}{8}$-inch bolts.

Spacing $= 7 \times 1.125'' = 7.875''$; $\frac{42 + 19}{7.875} = 8$ (approximately).

$e = \frac{5}{2} + 0.5 + \frac{1.125}{2} = 3.56''$; $g = 1.125 \times 1.5 + 0.75 = 2.43''$;

$\frac{3.56 + 2.43}{2.43} \times \frac{8{,}650}{8} = 2{,}670$ pounds. From Table IV., a bolt $1\frac{1}{8}$ inch in diameter can carry 2,670 pounds; therefore, use $1\frac{1}{8}$-inch bolts.

Use four bolts for attaching slipper to crosshead block. $G = 17{,}250$ pounds; $\frac{17{,}250}{4} = 4{,}325$. From Table IV., bolts should be $1\frac{1}{8}$ inch diameter.

Engine calculations should be tabulated in some systematic way so that they can be readily compared one with another. Fig. 27 shows an abbreviation of the form used by the Bureau of Steam Engineering, United States navy, and when results are tabulated in this way, attempts to lighten construction can be made in an intelligent manner by progressively decreasing the factor of safety of different parts, as experience shows this to be permissible. It will be noticed, in the results given, that the factor of safety varies from 9 to 19 for the middle of the piston rod. It would not seem that the factor of safety need be as great as 19 if rods have worked satisfactorily with a factor of 12 or 15. The quantities J, R_2 and N, given in Fig. 27, are portions of equation (21) in a slightly different form. The factor of safety obtained from the expression

$$N = \frac{A \times C}{W \times R}$$

should be the same as the factor of safety N used in equation 21. In the expression for R, π^2 is assumed to be 10, and E is assumed to be 30,000,000.

Connecting Rod.—The body of the connecting rod is figured as a column hinged at both ends, and the formulæ have the

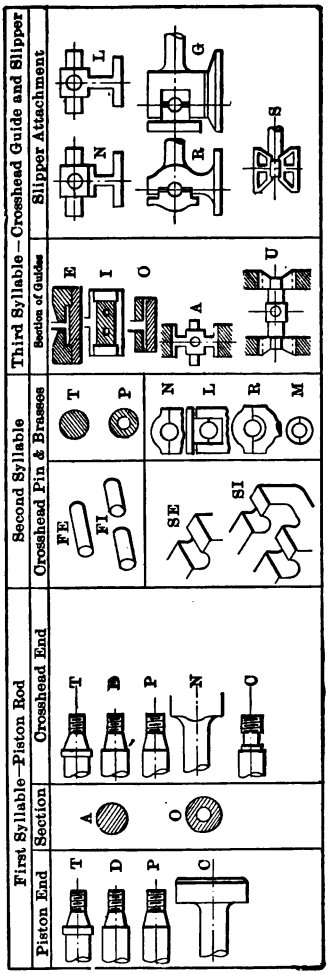


FIG. 27. DETAILS OF PISTON ROD, CROSSHEAD AND GUIDE.

Number.	Type.	Designed Load $W = \frac{2 \times \text{I.H.P.} \times 33{,}000}{P\,S}$.
1	Destroyer.....	76,000
2	Torpedo boat.....	33,500
3	Gunboat.....	65,300
4	Revenue cutter...	74,250
5	Battleship.....	295,000
6	Freighter.....	55,400
7	Freighter.....	78,400
8	Passenger boat...	99,000
9	Passenger boat...	78,500
10	Atlantic liner.....	164,000
11	Pacific liner.....	175,000
12	Pacific liner.....	171,500
13	Atlantic liner.....	339,000

Number.	Description.	Factor of Safety $N = \frac{A\,C}{W\,R}$
1	TON-SEL-ER...	9.05
2	TON-SEL-ER...	11.4
3	DON-SEL-EG..	15.3
4	TAP-FIT-EN...	17.8
5	TOC-FIP-EN...	10.8
6	DAD-FIT-EN...	18.2
7	DAD-FIT-EN...	17.75
8	TAD-FIT-AN...	12 2
9	DAD-FIT-EN...	19.2
10	CAD-FIT-EN...	15.8
11	TAP-FIT-EN...	18.5
12	TAD-FIT-US...	16.
13	TAD-FIT-US...	16.7

Diameter of Rod	Length of Rod l.	Radius of Gyration G.	$J = \frac{2l}{3G}$	Specified Ultimate Strength C.	Ratio Load at Middle to Steam Load $R = 1 + \frac{J^2C}{\pi^2E}$.	$\frac{\text{Length Conn. Rod}}{\text{Crank}}$ $\frac{l}{r}$
4½–2⅞	39 1/16	1.265	20.59	95,000	1.134	4.25
3 –1 13/16	24 1/16	0.876	22.12	95,000	1.155	4.44
4½–2	47⅛	1.23	25.6	95,000	1.208	4.46
5	50¼	1.25	26.85	80,000	1.192	5.
8 –3⅜	81	2.195	24.7	95,000	1.194	4.
5	66⅜	1.25	35.5	70,000	1.241	4.57
6	74 5/16	1.5	33.1	60,000	1.219	4.5
5½	65¼	1.32	33.2	70,000	1.258	4.5
5½	64	1.44	29.6	70,000	1.205	4.5
7½	82	1.88	29.1	70,000	1.198	4.44
8	91	2.	30.3	80,000	1.245	4.42
8	104¼	2.	34.7	70,000	1.28	4.18
11	114	2.75	27.6	70,000	1.176	4.

Factor of Safety at Root of Thread.		Cotan Sec $\frac{l}{r} = \frac{H}{W}$.	Load on Cross-head Guide H	Ahead.		Backing.	
Piston End.	Crank End.			Dimensions of Slipper.	Pressure per Square Inch.	Dimensions of Slipper.	Pressure per Square Inch.
8.8	7.65	0.242	18,400	11 x16	104.	2x 3 x16	211.
10.4	11.6	.231	7,740	7½x13½	76.5	2x 2 x13½	143.
10.4	9.6	.23	15,200	12 x14½	87.5	2x 4½x14½	108.
10.8	10.8	.204	15,150	13 x16	73.	2x 5 x16	95.
7.15	7.15	.258	76,200	23 x30	110.	2x 8½x30	149.
12.2	12.2	.224	12,400	12 x19	54.5	2x 4½x19	73.
11.8	11.8	.228	17,900	14 x22½	57.	2x 4½x22½	89.
10.9	10.9	.228	22,600	12 x24	78.2	12 x24	78.2
15.3	15.3	.228	17,900	16 x24	46.6	2x 6 x16	99.4
12.7	12.7	.231	37,900	22½x30	56.1	2x 8½x30	75.2
11.5	11.5	.232	40,650	24 x30	56.5	2x 8 x30	85.
10.3	10.3	.246	42,200	2x28x11	68.5	2x28 x11	68.5
11.3	11.3	.258	87,700	2x20x47	46.7	2x20 x47	46.7

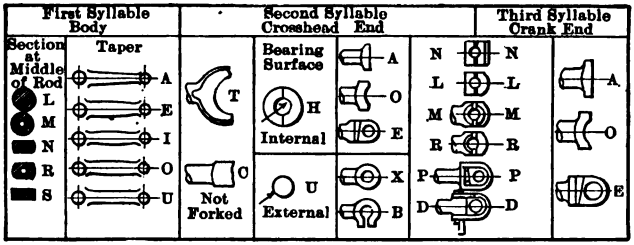


FIG. 28. CONNECTING ROD DETAILS. (SEE NEXT PAGE.)

same form as those for the piston rod, except that the constant is 1.08 instead of 0.48, and the factor of safety used is different. The middle portion of the rod is usually made tapered, and the formulæ give the diameter at the middle of the rod.

Solid rod:

$$D^2 = \sqrt{\frac{1.08 \times F \times C \times l^2}{10{,}000{,}000} + F^2} + F. \qquad (27)$$

Hollow rod:

$$D^2 = \sqrt{\frac{1.08 \times F \times C \times l^2}{10{,}000{,}000} + F^2 + (2F + d^2)d^2} + F. \qquad (28)$$

$F = \frac{2P}{\pi f}$; P = load upon connecting rod = $W \times k$; W = load on piston rod; l = length of connecting rod = $h \times$ stroke; h varies from 2 to 2½:

When h =	2	2.125	2.25	2.375	2.5
k =	1.033	1.03	1.026	1.023	1.021

f = allowable stress on rod, = $\frac{C}{N}$;

C = ultimate strength of material,
= 60,000 to 70,000 pounds for merchant engines,
= 80,000 to 95,000 pounds for naval engines;

N = factor of safety,
= 8 to 10 for naval engines,
= 12 for merchant engines;

Number.	Type.	Description.	Length of Rod, Center to Center, L.	$\frac{L}{r}$.	Cosec Sec—$\frac{L}{r}=K$.	Load on Rod $=Wk=P$.	Specified Ultimate Strength, C.	Ratio Load at Middle to Load at Ends $=R=1+\frac{L^2C}{G^{2.72}E}$
21	Destroyer.....	MA–TUX–MO...	46¼	4.25	1.03	78,200	95,000	1.489
22	Torpedo boat...	SAI–TUX–LA....	40	4.44	1.027	34,400	95,000	1.307
23	Gunboat.....	MA–TUX–NA....	67	4.465	1.026	67,000	95,000	1.957
24	Revenue cutter.	LA–THAN–NA..	75	5.	1.021	75,800	80,000	1.761
25	Battleship.....	MA–THAN–NA..	96	4.	1.032	304,200	95,000	1.495
26	Towboat.....	LA–THAL–LA...	65	5.	1.021	37,300	65,000	1.88
27	River boat.....	LA–TUX–NA....	68	4.535	1.026	42,200	95,000	1.845
28	Freighter.....	LA–THAN–LA..	94½	4.5	1.026	72,700	60,000	1.649
29	Passenger boat.	LA–THAN–NA..	94½	4.5	1.026	90,500	70,000	2.02
30	Atlantic liner...	LA–THAN–NA..	120	4.45	1.027	148,000	70,000	1.85
31	Pacific liner....	LA–THAN–LA..	138	4.18	1.031	176,800	70,000	2.

Number.	Smallest Section of Rod.				Section at Middle of Rod.					Factor of Safety at Middle of Rod $=N=\frac{AC}{PR}$
	Dimensions.	Shape.	Area.	Factor of Safety.	Dimensions.	Shape.	Area A.	Radius of Gyration G.	$\frac{L}{G}$.	
21	3½–2½	M	7.817	9.5	4.2 –2½	M	9.878	1.19	39.3	8.57
22	4.05x1½	S	5.063	13.9	4.45x2	S	8.9	1.285	31.1	18.8
23	4½–2	M	11.05	15.7	4.44–2	M	12.36	1.218	55.	8.95
24	5	L	19.64	20.8	5.62	L	24.8	1.405	53.3	14.9
25	7½–5	M	27.54	8.6	8½–5	M	35.43	2.44	39.3	7.4
26	3½	L	11.04	19.2	4.09	L	13.13	1.02	63.75	12.2
27	4	L	12.57	19.3	4.35	L	14.97	1.09	62.4	12.5
28	6½	L	30.68	25.4	6.64	L	34.55	1.66	57.	17.3
29	5	L	19.64	15.3	5¾	L	26.	1.44	65.6	10.
30	7½	L	41.4	19.6	8	L	50.2	2.	60.	12.8
31	7½	L	44.18	17.5	8½	L	56.75	2.125	65.	11.22

d = inside diameter of rod, and is usually from

$$\tfrac{1}{3}\sqrt{2F} \text{ to } \tfrac{2}{3}\sqrt{2F}.$$

The increase of the stress at the middle of the rod, due to the tendency of the rod to spring when being compressed, is generally about 75 percent (see column marked *R*, Fig. 28), so that theoretically the area at the very top of the column could be 57 percent of that at the middle and have no greater stress. The fork of the rod takes up about one-fourth of its length, so that the area at the smallest section should be about 80 percent of the area at the middle. In order that the area may be 80 percent, the diameter must be 90 percent. Due to the swinging of the rod across the line of the dead centers, and its being brought to rest on either side of the crank pin, there is bending introduced near the crank-pin box from the inertia of the rod. This additional bending is seldom figured, but the diameter here is increased as much as the diameter at the upper end is decreased, so that the rod has a uniform taper from end to end. The diameter at *G*, Fig. 29, should be 0.9 *H*, and the diameter at *J* should be 1.1 *H*.

The distance *Z* between the flat faces of the fork is made equal to *G*, or slightly greater. The heads of the cap bolts of the boxes of the fork must clear these faces by about ⅛ inch, and the diameter of the cross-head pin should be such that it clears the body of these bolts by about ¼ or ⅜ inch. It may be necessary to change the diameter of the cross-head pin after finding the diameter of these bolts, in order that there may be not too much nor too little clearance here.

The four bolts at the cross-head end, and the two at the crank-pin end, of the rod should be designed to carry the load *P*, and their diameters can be taken from Table IV. The nuts for the bolts are usually of the collar type, shown in Fig. 5 and Table V. The boxes at the forked end are made with *E* greater than *K* (see Fig. 29); the amount of the overhang to

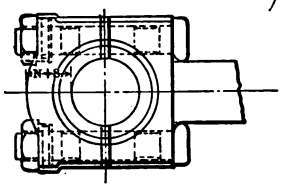


FIG. 29.

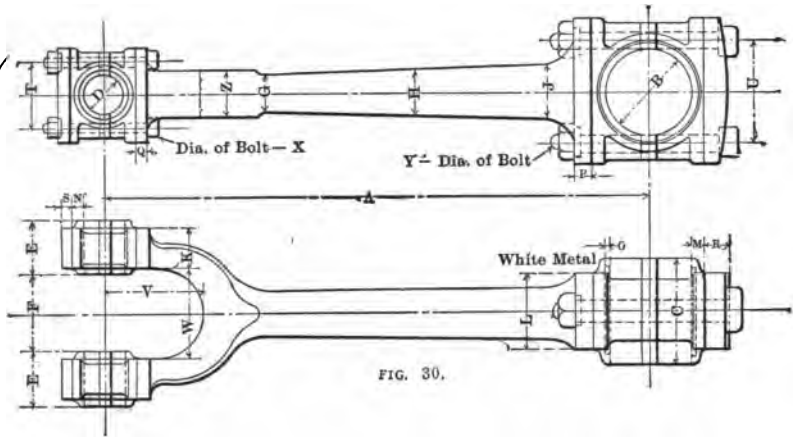


FIG. 30.

CONNECTING ROD AND DETAILS.

No.	A / Crank.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z
1	4.8	72″	9″	8⅞″	4½″	4½″	7″	3½″	4½″	4½″	3½″	7″	1½″	1″	½″	3″	2¼″	3″	2¼″	6⅞″	12⅜	10	8	1½	2½	3½″
2	5.	60	9	10⅛	4½	4½	7½	3⅝	4 3/16	4½	3¼	8	1⅛	....	½	3	2	2¼	N + S 2¼	6⅞	12¼	11	9	1½	2½	3⅝
3	....	54	10½	9⅛	5	5½	7⅛	4	4⅛	4¼	4¼	7½	1½	⅛	⅛	3⅛	2½	3⅛	2⅛	7½	14	11	8⅛	2	2½	4¼
4	....	60	9	9	4½	5	7½	4	4½	5	4	6¼	1⅛	....	....	3⅛	2½	2¼	N + S 2¼	7⅛	12⅛	10	8½	2	2½	4
5	4.5	81	12¾	15	7⅛	8½	10¼	5⅛	6	6½	6½	10½	2⅛	1½	½	4	3½	..	..	....	....	....	....	3½	4	....
6	4.	84	15	16¼	7½	9	13	6⅛	7½	8½	7½	12	2⅛	1½	½	....	3½	4½ R + M ¼	N + S ¼	11½	20½	12½	15½	3	4¼	6¼
7	4.18	138	20½	18⅜	10½	9⅞	11⅛	7½	8½	9½	7¼	13½	2¼	1⅛	¼	5½	4	8¼	4¼	14¾	26½	20⅛	13⅜	3¾	5	8

be allowed in any case being determined by comparison with a similar rod in the table accompanying Fig. 29. F should be taken from the computation for the cross-head block, allowance being made for the facing strip and a slight clearance, 1/32 inch or less. $W = F + E - K$, E being equal to the length of the cross-head pin. V should be taken from the details of the cross-head block, the bottom of the fork clearing the end of the piston rod nut by about the thickness of the nut. The thickness Q should be from ¼ to ½ inch greater than X.

The thickness S of the caps is determined by considering the cap as a beam of breadth K, supported at the ends by the cap bolts; so that the span is T. As the load is distributed over a part of the beam, the bending moment is generally obtained from the formula

$$M = \frac{W \times l}{6}, \text{ where } l = T, \text{ and } W = \frac{P}{2}; \text{ thus } M = \frac{P \times T}{2 \times 6};$$

$$I = \frac{K \times S^3}{12}; f = \frac{M \times y}{I} = \frac{P \times T}{2K \times S^2};$$

then

$$S = \sqrt{\frac{P \times T}{2 \times K \times f}}. \qquad (29)$$

Where P = load upon connecting rod;
T = distance between center lines of cap bolts;
K = breadth of cap;
f = allowable stress.

The load upon the caps is intermittent in character, so that the factor of safety should be 8. The caps are usually made of bronze, cast steel or wrought steel. The thickness N of the brasses can be taken from the table of Fig. 29. The thickness of the metal of the caps, where they inclose the bolts, is generally about $\frac{X}{4}$. Between the upper and lower brasses there

is one thick cast iron liner, usually from 1 inch to 1¾ inches in thickness, held in by dowel pins and several thin tin liners varying from 1/16 to 1/64 inch.

The fork of the connecting rod is figured as if the ends were free to move. The force $\frac{P}{2}$, considered as acting at the middle of the length of the box, will cause bending, direct

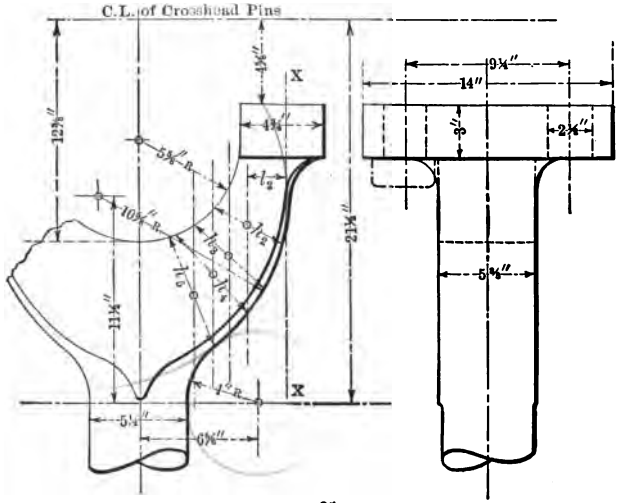


FIG. 31.

compression and shear upon a given section of the fork. It is usual to consider only the bending, and to allow for the direct compression and shear by using a low stress. The design can be simplified by proceeding as in Fig. 31. The inside of the fork is semi-circular in shape, the semi-circle at its lowest point being a distance V from the center line of the cross-head pins, and the radius of the semi-circle being equal to $\frac{W}{2}$, see Fig. 29.

To get the outline of the outside of the fork, draw a number of lines a distance l apart, parallel to $X\ X$, which passes through the center of the length of the bearing. Certain sections of the fork lying about normal to its outside edge will have their neutral axes upon these lines. The breadth of the sections will be Z (see Fig. 29), and the length must be sufficient to cause the stress upon them to be within the desired limits. The bending moment upon the section l_1'' distant from $X\ X$ will be $\frac{P \times l_1}{2}$. If h is the length of the section, then

$$f = \frac{P \times l_1 \times h_1 \times 12}{2 \times 2 \times Z \times h^2} = \frac{3Pl_1}{Z \times h^2}$$

or,
$$h_1^2 = \frac{3 \times P \times l_1}{Z \times f}. \qquad (30)$$

Since, for the other sections, everything else is the same except the distance l_1, and since

$$l_2 = 2l_1, \quad l_3 = 3l_1, \quad l_4 = 4l_1, \text{ etc.},$$

then
$$h_2^2 = 2h_1^2,\ h_3^2 = 3h_1^2,\ h_4^2 = 4h_1^2, \text{ etc.}$$

The distances h_1, h_2, h_3, etc., can be laid off about normal to the semi-circle, so that their centers fall on the lines distant l_1'', l_2'', l_3'', etc., from $X\ X$. Through the ends of these lines the edge of the other side of the fork can be drawn. The curves should be arcs of circles, for the sake of convenience. Sometimes the construction of the piece is simplified by substituting a straight line for the curve. In this case the line must pass outside of all the points laid off, and for this reason will give a heavier fork than before.

The crank-pin end of the rod cannot be completed until the size of the crank pin is determined, and this depends upon the size of the crank shaft. When the size of the crank shaft has

been determined, and from that the size of the crank pin, the length C (see Fig. 29) of the box should be such that the bearing pressure per square inch does not exceed that given in Table VII. The load upon the crank pin is the resultant of the thrust through the rod, due to the steam pressure on the piston, the inertia and weight of the reciprocating parts, and the centrifugal force of the crank-pin end of the connecting rod. The following formula gives the approximate mean load upon the crank pin:

$$L = 1.6 \times \frac{21{,}000}{P.S.} \times I.H.P., \qquad (31)$$

where $I.H.P.$ = the indicated horsepower of the cylinder over the pin;

$P.S.$ = the piston speed of the engine in feet per minute.

The factor, 1.6, may vary from 1.4 to 1.8, depending upon the distribution of power among the cylinders, as there would be considerable bearing pressure upon the pin, due to the inertia of the reciprocating parts and the centrifugal force of the end of the connecting rod, even if there were very little power being developed in the cylinder. The factor 1.6 is for average conditions, and should be increased if the cylinder develops less than its share of the power, and decreased if more is developed.

The cap of the crank-pin end is designed to carry the total load P, so that the formula for the thickness R becomes

$$R = \sqrt{\frac{PU}{Lf}}, \qquad (32)$$

where P = load upon connecting rod (by Formula 20);

U = distance between center lines of bolts;

L = breadth of cap;

f = stress to give a factor of safety of 8.

The bolts, brasses, etc., are obtained in the same way as for

the forked end of the rod. The two parts of the box are separated by liners, a cast iron liner whose thickness varies from 2½ to 4½ inches, and thinner brass and tin liners. The cap may be of wrought steel, cast steel or bronze. When made of cast steel or bronze the shape is generally that shown in Fig. 30. The bolts should clear the crank pin by an amount varying from ⅜ inch to ⅝ inch.

Calculations for the Connecting Rod.

Let $l = 2.25 \times \text{stroke} = 2.25 \times 42'' = 94.5''$

$k = 1.026;$

$P = 77{,}700 \times 1.026 = 79{,}800$ pounds;

$N = 12;$

$C = 70{,}000$ pounds;

$$f = \frac{70{,}000}{12} = 5{,}825 \text{ pounds.}$$

$$F = \frac{2 \times 79{,}800}{3.142 \times 5{,}825} = 8.73.$$

Formula (27):

$$D^2 = \sqrt{\frac{1.08 \times 8.73 \times 70{,}000 \times (94.5)^2}{10{,}000{,}000}} + 76.2 + 8.73 = 34.53.$$

$D = 5.88''$, (Use 5⅞ inches), diameter at middle of rod.

Diameter at top of rod $= 5.875 \times 0.9 = 5.28$, (Use 5¼ inches);

Diameter at bottom of rod $= 5.875 \times 1.1 = 6.45$, (Use 6½ inches).

Load on bolts of crosshead pin caps $= \frac{79{,}800}{4} = 19{,}950$ pounds; this will require 2.5-inch bolts (see Table IV.).

$$T = 6.25'' + 2.5'' + 0.5'' = 9.25''.$$

Heads of bolts will be $1.5 \times 2.5'' = 3.75''$ diameter.

$9.25'' - 3.75'' = 5.5''$. The thickness Z of the fork can be 5⅜ inches and have the heads of the bolts clear the fork by $1/16$ inch.

The length of the crosshead pin = 6.25 inches. Let the overhang of the box be $\frac{3}{4}$ inch on each side; $K = 6.25'' - 1.5'' = 4.75''$. We will make the caps of wrought steel.

Formula (29): $$S = \sqrt{\frac{79{,}800 \times 9.25}{2 \times 4.75 \times 7{,}500}} = 3.22''$$ (Use $3\frac{1}{4}''$).

$$F = 9.5'' + .25'' = 9.75'';$$
$$W = 9.75'' + 1.5'' = 11.25'';$$
$$V = 4.375'' + 4'' + 4'' = 12.375'';$$
$$Q = 2.5'' + .5'' = 3'';$$
$$N = 1.5''.$$

The radius of the inside of the fork is $\frac{W}{2} = 5.625$ inches.

Draw lines parallel to XX, Figure 31, spaced 1 inch apart.

Formula (30): $$h^2 = \frac{3 \times 79{,}800 \times 1}{5.375 \times 5{,}000} = 8.92; \quad h = 2.985 \text{ inches.}$$

$h^2_2 = 2 \times 8.92 = 17.84$; $h_2 = 4.23$ inches.
$h^2_3 = 3 \times 8.92 = 26.76$; $h_3 = 5.18$ inches.
$h^2_4 = 4 \times 8.92 = 35.68$; $h_4 = 5.97$ inches.
$h^2_5 = 5 \times 8.92 = 44.60$; $h_5 = 6.68$ inches.

The dimensions of the collar nuts can be taken from Table V.; total thickness of liners for crosshead end to be 2 inches, as follows:

1 cast iron liner	$1\frac{5}{8}$	inches thick,
1 brass liner	$\frac{1}{4}$	inch thick.
1 tin liner	$1/16$	inch thick.
1 tin liner	$1/32$	inch thick.
2 tin liners	$1/64$	inch thick.

Each bolt of the crank pin box must carry $\frac{79{,}800}{2} = 39{,}900$ pounds.

From Table IV., the diameter should be $3\frac{1}{4}$ inches.

Formula 37 (see below under crank shafts) gives the size of the crank shaft.

$$D = (0.038 \times 23.5 + 0.009 \times 41 + 0.002 \times 64. + 0.0165 \times 42.) \times \sqrt[3]{185}$$
$$= 11.9'', \text{ (Use 12 inches).}$$

Make the diameter of the crank pin $12\frac{1}{2}$ inches.

Formula (31): $L = 1.6 \times \dfrac{21{,}000 \times 1{,}000}{850} = 39{,}500$ pounds.

From Table VII., allowable pressure = 225 pounds. $\dfrac{39{,}500}{225} = 175.5$ square inches. Length C of crank pin $= \dfrac{175.5}{12.5} = 14$ inches. Let the box overhang 2 inches on each side; then $L = 10$ inches. $U = 12.5'' + 3.25'' + 0.75'' = 16.5''$. Make the cap of cast steel.

Formula (32): $R = \sqrt{\dfrac{79{,}800 \times 16.5}{10 \times 5{,}000}} = 5.12''$, (Use $5\frac{1}{4}$ inches).

Let $M = 2''$, $O = 0.625''$, $P = 3.75''$. Total thickness of liners for crank-pin end to be $3\frac{1}{2}$ inches, as follows:

1	cast iron liner	$2\frac{3}{4}$ inches thick.
1	brass liner	$^9/_{16}$ inch thick.
2	tin liners	$^1/_{16}$ inch thick.
1	tin liner	$^1/_{32}$ inch thick.
2	tin liners	$^1/_{64}$ inch thick.

SHAFTING.

Crank Shaft.—The stress upon the crank shaft can be figured by means of the formula $f = \dfrac{My}{I}$,

where M = the maximum equivalent twisting moment;
y = radius of shaft;
and I = polar moment of inertia.

or $$f = \frac{TD}{\frac{4\pi D^4}{64}} = \frac{16T}{\pi D^3}.$$

For solid shafts, $$D = \sqrt[3]{\frac{T \times 5.1}{f}}. \quad (33)$$

For hollow shafts, $$f = \frac{T \times D \times 5.1}{D^4 - d^4}. \quad (34)$$

The maximum twisting moment can be obtained from the mean twisting moment by means of a factor derived from analyses of engine trials. The mean twisting moment

$$= \frac{I.H.P. \times 33{,}000}{2\pi n}, \quad (35)$$

where $I.H.P.$ = total indicated horsepower of engine,
and n = revolutions of engine per minute.

The maximum can be obtained from this by multiplying by a factor C, which has the following values:

Single-cylinder engine,	$C = 2.$
Two-cylinder engine,	$C = 1.5$
Three-cylinder engine,	$C = 1.33$
Four-cylinder engine,	$C = 1.25$

The crank shaft is subjected to bending as well as twisting, and the equivalent twisting moment is usually found by use of the formula:

$$T_1 = M + \sqrt{M^2 + T^2}, \quad (36)$$

where M = the bending moment at the section where
T = the maximum twisting moment.

The load upon the crank shaft is intermittent in character, varying from a maximum to a minimum. The factor of safety to be used, then, is 8 if the twisting moment is that given by

formula (36). If the bending moment is neglected, and only the twisting moment is used in figuring the shaft, the factor of safety should be increased to 10.

In order to get a rating it is necessary that not only a ship's hull but also certain parts of its machinery shall conform to the rules of the registering society. The crank shaft is one of the parts of machinery whose size is specified, so that its diameter will be determined by some of the following rules, rather than by the preceding formulæ.

Lloyd's Rules (1906-1907).—Stroke greater than ½ low-pressure diameter and less than low-pressure diameter.

Three Cranks at Equal Angles.—Ratio of low-pressure area to high-pressure area not over 9:

Diameter of line shaft

$$= (0.038A + 0.009B + 0.002D + 0.0165S) \times \sqrt[3]{P}. \quad (37)$$

Four Cranks at Equal Angles.—Ratio of low-pressure area to high-pressure area not over 12:

Diameter of line shaft

$$= (0.033A + 0.01B + 0.004C + 0.0013D + 0.0155S) \times \sqrt[3]{P}. \quad (38)$$

Diameter of crank shaft = 21/20 diameter of line shaft.

Diameter of thrust shaft between collars = 21/20 diameter of line shaft.

Diameter of screw shaft

$$= \left(0.63 + \frac{0.03P}{T}\right) \times T,$$

where P = diameter of propeller, and T = diameter of line shaft. In no case must the diameter of screw shaft be less than 1.07 T.

In the above,

A = diameter of high-pressure cylinder in inches;

B = diameter of first medium-pressure cylinder in inches;

C = diameter of second medium-pressure cylinder in inches;

D = diameter of low-pressure cylinder in inches;

S = stroke of engine in inches;
P = gage boiler pressure.

Bureau Veritas Rules (1907) for shafting of double, triple and quadruple expansion engines, no overhung cranks:

$$d = \sqrt[3]{\frac{PL(n_1 D_1^2 + 0.1nD^2)}{C}}, \qquad (39)$$

where d = diameter of after shaft bearing in inches;
n_1 = number of high-pressure cylinders;
D_1 = diameter of each high-pressure cylinder in inches;
n = number of low-pressure cylinders.
D = diameter of each low-pressure cylinder in inches.
L = length of stroke, in inches, common to all pistons.
P = gage boiler pressure in pounds per square inch.

For hollow shafts the diameter must be increased by 1 percent if the diameter of the hole is 0.4 of the outside diameter; by 2 percent if the diameter of the hole is 0.5 of the outside diameter; by 5 percent if the diameter of the hole is 0.6 of the outside diameter, and by 10 percent if the diameter of the hole is 0.7 of the outside diameter.

Type of Engine.	No. of Cylinders.	No. of Cranks.	Angle Between Cranks.	Value of Factor C.
Compound.....	2	2	90°	3,400
Compound.....	4	2	90°	3,500
Compound.....	6	3	120°	3,800
Compound.....	2	1	....	2,100*
Compound.....	3	3	120°	3,600
Triple.....	3	3	120°	3,900
Triple.....	3	2	90°	3,000
Triple.....	4	2	90°	3,300
Quadruple.....	4	2	90°	3,100†
Quadruple.....	4	4	90°	4,000†

* Cutoff at 0.8 in high-pressure cylinder.
† Subject to approval.

If the sequence of cylinders and angles of the cranks in quadruples are so chosen as to reduce the maximum torsional moment the above values may be increased, but in no case shall they exceed 4,100. Each particular case must be submitted for approval.

The diameter of propeller shaft must be $(1.7 \frac{D}{d} - 15)$ percent in excess of the diameter of the crank shaft; where D = diameter of propeller in inches, and d the diameter of crank shaft in inches. The diameter of thrust shaft at the bottom of collars, both between and immediately beyond these latter, to be equal to that of the crank shaft, and tapered off at each end to the smaller diameter of the body of the shaft. For tunnel shafts, a reduction of 6 percent on the diameter of the crank shaft will be allowed.

Turbine Shafting.—In turbine engines, where *I. H. P.* is the estimated power transmitted by each shaft,

$$d = \sqrt[3]{\frac{70 \times I.H.P.}{R}}; \quad d_1 = d + \frac{D}{160}; \quad d_2 = 1.05d, \qquad (40)$$

where d = diameter of tunnel shafting in inches;

d_1 = diameter of propeller shaft in inches;

d_2 = diameter of rotor shaft in inches at smaller part;

D = diameter of propeller in inches;

and R = number of revolutions per minute.

American Bureau of Shipping Rules for Shafting.

$$S = \sqrt[3]{\frac{PD^2L}{K\left[\frac{D^2}{d^2} + 2.4\right]}}, \qquad (41)$$

where S = diameter of crank shaft in inches;

P = gage pressure at boiler in pounds per square inch;

D = diameter of low-pressure cylinder in inches (if there is more than one low-pressure cylinder, then for D^2 use the sum of the squares of the diameters);

L = length of stroke in inches;

K = factor from following table:

Type of Engine.	No. of Cranks.	Angle Between Cranks.	Value of Factor K.
Compound.....	2	90°	1,450
Compound.....	2	180° or 0°	1,200
Compound (3 cylinders)...	3	120°	1,500
Triple.....	3	120°	1,700
Triple (4 cylinders).....	4	90°	1,600
Quadruple.....	4	90°	1,800
Quadruple (5 cylinders)....	5	72°	2,100

In four-crank engines with unequal spacing of cranks, K must be decreased as follows:

If smallest angle lies between 75 degrees and 85 degrees, decrease K by 100; if smallest angle lies between 65 degrees and 75 degrees, decrease K by 200; if smallest angle lies between 55 degrees and 65 degrees, decrease K by 300.

For hollow shafting see under equation 39. The diameter of the propeller shaft is to be 6 percent in excess of the diameter of the crank shaft. The diameter of the tunnel shafting may be 5 percent less than that of the crank shaft.

Crank shafts are generally made in sections, one section for each cylinder. The sections are flanged and joined together by coupling bolts. Shafts may be divided into two classes, forged and built-up, and still further divided into solid shafts and hollow shafts. In the merchant marine, shafts less than 12 inches diameter are usually forged, as shown in Fig. 32, except that they are seldom hollow; above that diameter they are usually built up, as shown by Fig. 33. Hollow shafts are

seldom used in merchant ships, but are used almost altogether in vessels of the navy.

The proportions of the different parts should be as follows:

Diameter	$= D$ (see above).
Thickness of webs	$= A = 0.6D$ to $0.7D$.
Length of crank pin	$= B =$ length of crank pin box + $\frac{1}{8}$ inch to $\frac{3}{8}$ inch.
Diameter of crank pin	$= C = D$ to $D + 1$ inch.
Thickness of couplings	$= E = 0.22D$ to $0.25D$, naval engines.
	$= 0.25D$ to $0.28D$, merchant engines.
Clearance between webs and main bearings	$= F = \frac{1}{4}$ inch to $\frac{3}{8}$ inch.
Clearance between couplings and main bearings	$= G = 2$ inches to 3 inches.
Clearance between eccentric pads and main bearings	$= H = \frac{1}{4}$ inch to $\frac{1}{2}$ inch (minimum).
Radius of coupling fillet	$= d = 1$ inch to 2 inches.
Diameter of eccentric pads	$= K = D + \frac{1}{2}$ inch to $D + \frac{3}{4}$ inch.
Diameter of journal and pin in crank webs	$= L = C + \frac{1}{2}$ inch to $C + 1$ inch.
Radius of crank web at pin	$= M = 0.875L$.
Radius of crank web at shaft	$= N = 0.925L$.
Breadth of web in solid shafts	$= O = 1.05D$ to $1.1D$.
Metal between shaft and pin	$= P = 0.45L$ (at least).

When the three sections of shafting for a triple engine are to be interchangeable, all of the crank webs should be of the same thickness, otherwise the forward webs can be made thinner than the after webs, having a thickness of 0.6 D at the forward end and 0.7 D at the after end. The bearings can be of different lengths, but sufficient clear space must be allowed on all sections of the shaft to accommodate the longest bearing, if the sections are to be interchangeable.

The diameter of the coupling flange must be determined by the size of the coupling bolts used. The pitch circle of the bolts must be large enough to permit the nuts on the ends of the bolts to turn, without having to cut away anything but the fillet of the flange. The correct number of bolts and diameter of pitch circle can be determined only by trial. In the first assumption, the radius of pitch circle can be taken as

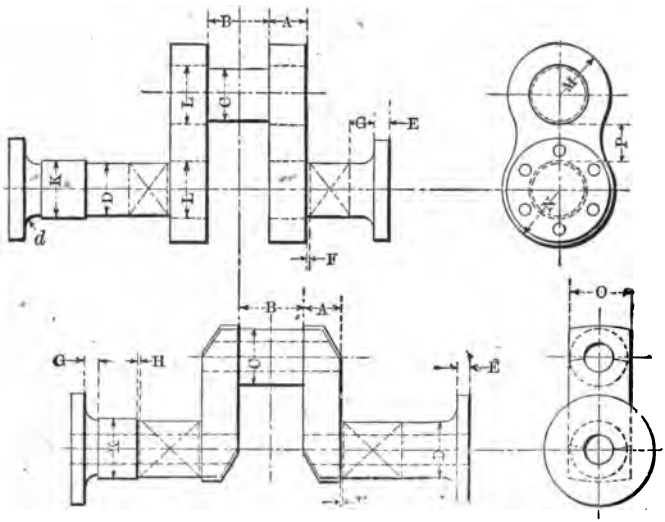


FIG. 32 (bottom). FIG. 33 (top).

0.75 of the diameter of the shaft, and in the largest sizes of shafts one bolt is allowed for about each 2 inches of diameter of shaft. In order that the shearing strength of the bolts may equal the torsional strength of the shaft, the diameter of the coupling bolt should be, for solid shafts,

$$R = \frac{D}{2} \times \sqrt{\frac{D}{n \times r}}, \qquad (42)$$

and for hollow shafts:

$$R = \tfrac{1}{2}\sqrt{\frac{D^4 - d^4}{D \times n \times r}}, \qquad (43)$$

where R = diameter of coupling bolt at shearing section;

D = outer diameter of crank shaft;

d = inner diameter of hollow shaft;

n = number of coupling bolts;

and r = radius of pitch circle of coupling bolts.

The diameter of the coupling flange should be $2(r + R)$.

The coupling bolts are usually tapered from end to end at the rate of 1 inch per foot. Where the nut is attached at the smaller end the diameter of the bolt can be reduced from ¼ inch to ½ inch, as the nut serves merely to keep the bolt in place.

BEARINGS.

The diameter of the crank shaft in the main bearings is that given by the formula for crank shafts. The length of the bearings is made such that the bearing pressure shall not exceed that given in Table VII. The loads upon the bearings increase as we go towards the propeller, and are not all of the same character. In some cases the bearing pressure is well distributed over the entire circumference of the bearing; in other cases it tends to act upon a small portion only. As the important thing about a bearing is that it shall be sufficiently large to be kept cool, it can be seen that a larger unit bearing pressure can be allowed where the load is well distributed than where it is concentrated upon one part only.

When a piece of shafting, such as a line shaft, for instance, is transmitting power, there is no load upon the supports except that due to the weight of the shafting. If, however, a crank is introduced into the shaft, and a bearing placed on either side of the crank, these bearings will be subjected to

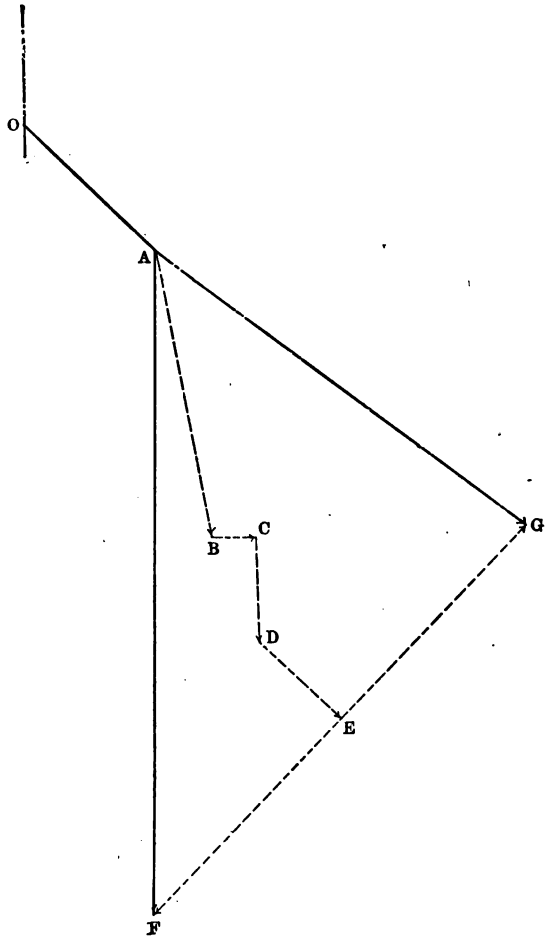


FIG. 34.

loads in addition to those from weight alone. The force which one web delivers through the crank pin to the other web will be felt upon the bearing adjacent to the latter web,

while the reaction which the first web experiences will be felt upon the bearing nearest that web. These loads upon the bearings must be equal in amount and opposite in direction, in order that the shaft which was in equilibrium before the crank was introduced may still be in equilibrium.

The turning force acting at the crank pin will increase as we go towards the propeller, so that this component of the bearing pressure will become more predominant. The components of the bearing pressure are shown in Fig. 34. *OA* is the position of the crank after it has turned through 135 degrees. *AB* is the force acting through the connecting rod, and is composed of the weight of the reciprocating parts and the unbalanced steam pressure on the piston. *BC* is the cross-throw of the connecting rod. *CD* is the resultant of the weight of the shaft and the inertia of the reciprocating parts. *DE* is the centrifugal force of the unbalanced parts of the crank webs and crank pin. *EF* is the force acting upon the after web, due to the turning force from the forward cylinders. *EG* is the reaction upon the forward web, due to the above force. *AF* is the resultant force acting upon the after bearing, and *AG* is the resultant for the forward bearing.

It will be seen that, if the turning force were left out, as is the case in the bearings of the first cylinder, the two bearings would have the same load upon them, and acting in the same direction. The effect of the turning force, however, is to cause the resultant loads to differ quite widely in amount and in direction. As the turning force changes its direction of action through 360 degrees during a revolution, its effect upon the resultant, as it becomes more predominant in the bearings nearer the propeller, is to cause the load to be distributed more uniformly around the bearing. In the case of the forward bearing of each pair of bearings, since the reaction of the turning force is, in general, acting in a direction opposite

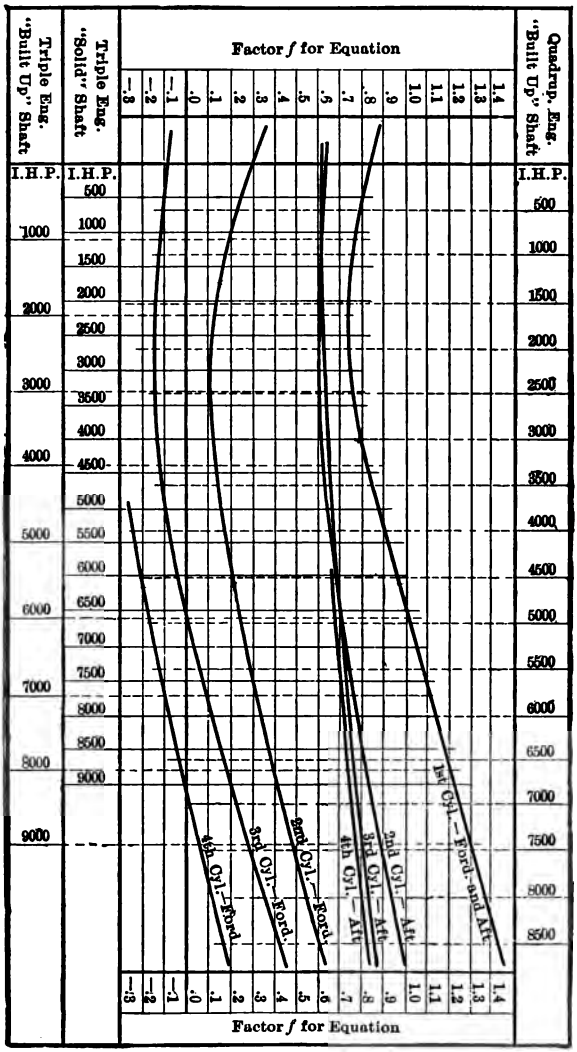

Triple Exp. "Built Up" Shaft
Triple Exp. "Solid" Shaft
Factor f for Equation
Quadrup. Exp. "Built Up" Shaft
I.H.P.
500
1000
1500
2000
2500
3000
3500
4000
4500
5000
5500
6000
6500
7000
7500
8000
8500
9000
−.3
−.2
−.1
.0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
1.1
1.2
1.3
1.4
1st Cyl.—Ford. and Aft.
2nd Cyl.—Aft.
3rd Cyl.—Aft.
4th Cyl.—Aft.
2nd Cyl.—Ford.
3rd Cyl.—Ford.
4th Cyl.—Ford.

FIG. 35.

to that of the other forces, the resultant tends to act upon the sides of the bearings.

The mean load upon the bearings can be found approximately by means of the formula:

$$L = \frac{21{,}000}{PS}[HP_t + fHP_c], \qquad (44)$$

where PS = piston speed in feet per minute;

HP_t = the indicated horsepower developed by the cylinders forward of the cylinder whose bearings are in question;

HP_c = the indicated horsepower deve.oped in the cylinder over the bearings;

and f = a factor whose value can be taken from the curves given in Figure 35.

The factor f allows for two components of the bearing load, one due to the power being developed in the cylinder over the bearings, and the other due to the centrifugal force of the rotating parts. If we had to deal with the first component only, f would be constant for all horsepowers, but since the centrifugal force of the rotating parts increases at a greater rate than the horsepower, the factor f will increase as the horsepower increases. When the centrifugal force of the rotary parts is balanced by weights upon the crank webs, the factor f will be taken as for zero *I. H. P.*

Sometimes the bearings between two cylinders are combined into a common bearing. In this case the load coming upon it can be found approximately by calculating the load coming upon each bearing by means of formula 44, and multiplying the sum of the two loads by a factor taken from the following table:

TABLE OF FACTORS FOR COMMON BEARINGS.

Between.	Type of Engine.	Angle Between Cranks.	Factor.
1st and 2d cylinders..	Triple; high leading....	120°	0.6
	Quadruple.....	180°	0.45
	Triple; low leading.....	240°	0.85
2d and 3d cylinders..	Quadruple.....	90°	0.47
	Triple; high leading....	120°	0.625
	Triple; low leading.....	240°	0.9
3d and 4th cylinders..	Quadruple.....	180°	0.875

When two cranks are at 180 degrees from one another, with a common bearing between them, the steam loads are opposed to one another, and balance, if equal powers are developed in the two cylinders. On the other hand, the loads from the turning force, being transmitted through the crank pins, are added to one another. This can be readily seen when it is remembered that the common bearing serves as the after bearing of one cylinder, and the forward bearing of the other cylinder. Since the cranks make an angle of 180 degrees, the two forces, which in the bearings for a single crank are opposed to one another, are in this case added to one another.

It will be found usually that in the case of triple-expansion engines the two bearings of the first cylinder and the forward bearing of the second cylinder can be made of the same size, also that the after bearing of the second cylinder and the forward bearing of the third cylinder can be made the same. The after bearing of the third cylinder will be the largest of all.

Engine Bed.—The stresses to which the engine bed will be subjected are very hard to calculate accurately, as they depend upon the alinement, balance, etc. The thickness of metal necessary for a rigid bed is determined more by experience, and is usually from ⅞ inch to 1½ inches when the bed is of

cast iron, and from ¾ inch to 1¼ inches when of cast steel. When made of cast iron, the longitudinal girders and cross girders carrying the main bearings are made of the hollow box form (see Fig. 36), usually open at the bottom, with

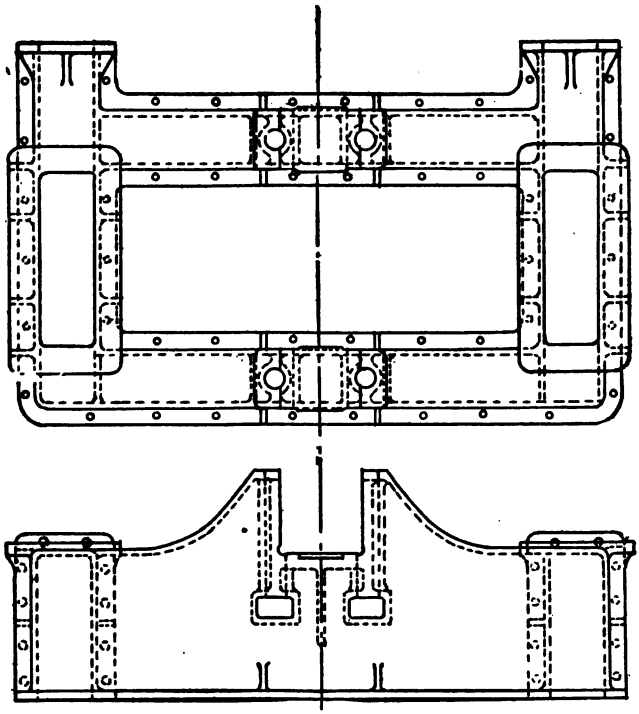


FIG. 36.

flanges for the holding-down bolts. When made of cast steel, it is found best to make all parts of the open *I* section, a section consisting of a central web with broad flanges at the top and bottom. (See Figs. 37 and 38.)

The bed is usually made in sections, to facilitate casting and handling, the break in the bed usually coming opposite the.

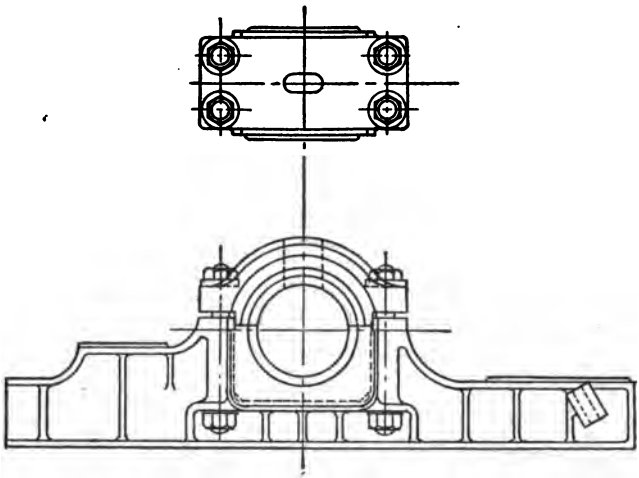


FIG. 37.

coupling flanges of the shaft. The cross girders can be figured as beams supported at the center line of the longitudinal girders, and loaded in the center of their length with the load for which the bearing caps are figured. The width of the cross girders is less than the length of the bearings which

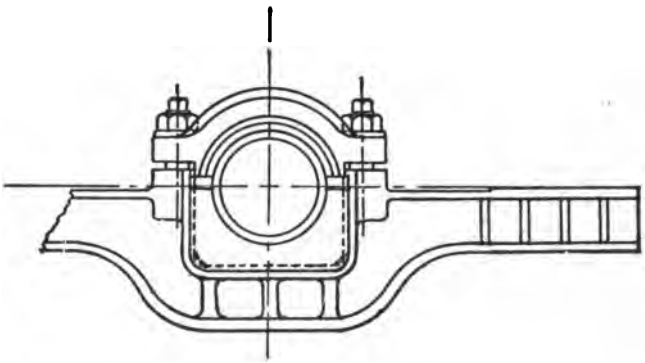


FIG. 38.

they carry, the amount of the overhang of the bearings being about three-fourths of the overhang of the boxes of the crank-pin end of the connecting rod. The sectional area of the cross girder under the bearing should be sufficient to keep the stress from 600 pounds to 1,000 pounds per square inch for cast iron, and from 1,000 pounds to 1,500 pounds for cast steel. As the after bearing is subjected to the greatest load, the girder carrying it should be figured, and the depth of the other girders taken from this. This depth will be the minimum depth; that taken finally may be such as will cause the connecting rod to clear the sole plate at the bottom of the stroke.

The distance between the two longitudinal girders should be sufficient to clear the connecting rod as it rotates. To do this it is best to draw a clearance diagram for the crank-pin end of the connecting rod, showing the path of the extreme points. The side girders should then clear this path by 1 inch to 2 inches. Usually the cross girders are carried from 1½ to 2 inches below the lowest point of the clearance diagram, otherwise the sole plate must be cut away at the crank pits to clear the connecting rod.

Main Bearing Caps.—The main bearing caps are made of cast iron, cast steel, wrought steel or bronze. When made of wrought steel, the brasses are flat backed, as shown in Fig. 39. When made of cast iron, cast steel or bronze, they are of the shape shown in Fig. 37, and the white metal is put into recesses cored out in the cap. The caps are always provided with hand holes, by means of which the temperature of the shaft can be ascertained; they are also used for applying grease to the shaft. The hand holes are 2½ inches wide and 4½ or 5 inches long, the narrower dimensions being in the direction of the center line of the shaft. The width of the cap will be the same as that of the cross girder carrying the bearing, or from 1 inch to 3½ inches less than the length of

the bearing. The load to be used in figuring the bolts and cap should be the maximum upward load coming upon the cap. In the after bearings of the different cylinders the greater loads act upon the top and bottom of the bearings, while in

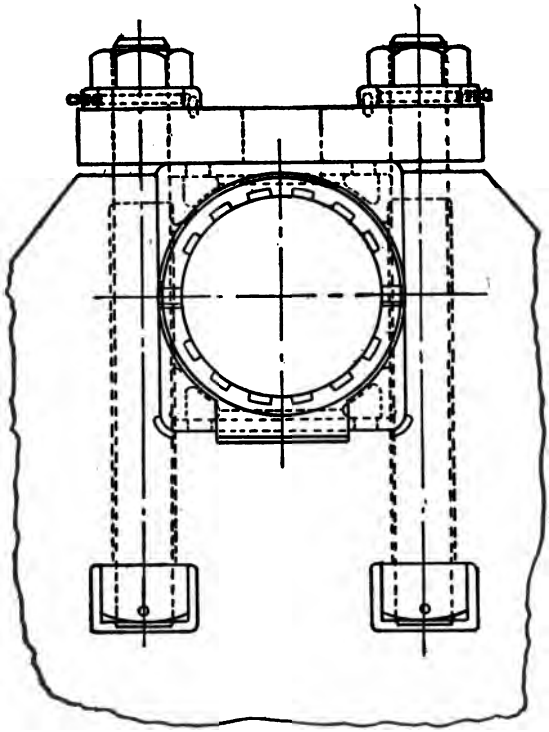


FIG. 39.

the forward bearings they act upon the sides of the bearings. Caps will be figured, however, as if the maximum loads acted upon the tops of all the bearings. The following factor, applied to the mean loads obtained from formula 44, will give the maximum loads coming upon the bearing caps:

Cylinder.	Type of Engine.	Which Bearing.	Factor.
1st.....	All.....	Forward and after..	2.
2d.....	Triple.....	Forward and after..	1.75
2d and 3d.....	Quadruple.....	Forward and after..	1.75
3d.....	Triple.....	Forward.....	1.67
4th.....	Quadruple.....	Forward.....	1.67
3d.....	Triple.....	After.....	1.4
4th.....	Quadruple.....	After.....	1.4

The size of the bolts to carry these loads can be taken from Table IV. The number of bolts in the caps depends upon the construction of the bed. If the bed is of cast iron, two bolts are usually put into each bearing to hold down the cap (see Fig. 36), unless the cap is long, when four bolts will be used. If the bed is made of cast steel, and the cross girders are of the open I section, four bolts are commonly used in order to clear the central web (see Fig. 37). Sometimes a boss is cast in the central web, and two collar studs used, as shown in Fig. 38. When the construction shown in Fig. 36 is used, the studs can be placed close to the shaft, allowing a clearance of ½ to ¾ inch. When the construction is as shown in Figs. 37 and 38, the clearance will be much more.

The caps should be figured as beams supported at the ends and the load distributed over a part of the length. Using the maximum load for which the bolts were designed, the bending moment will be $\frac{W l}{6}$, where W = load and l = distance between the center lines of the bolts. In obtaining the moment of inertia of the section of the cap, the breadth to be used should be the breadth of the cap minus the breadth of the hand hole, usually 2½ inches.

The height of the cap will be:

$$H = \sqrt{\frac{Wl}{fB}}, \qquad (45)$$

where W = the maximum load upon the cap;
l = the distance between the center lines of bolts;
B = the net breadth of cap;
and f = such a stress as will give a factor of safety of 10, since the caps may be subjected to pounding.

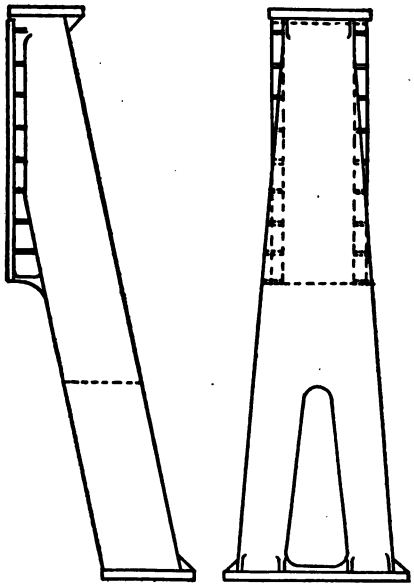


FIG. 40.

If the top of the cap is cored out, the formula

$$\frac{BH^3 - bh^3}{12}$$

should be used in getting the moment of inertia of the section, where b is the breadth and h is the height of the cored space. In this case the cap will be figured from the formula:

$$f = \frac{WlH}{BH^3 - bh^3}. \qquad (46)$$

ENGINE FRAME.

The engine framing is made of cast iron, cast steel and wrought steel. When made of cast iron, the framing usually consists of hollow inverted Y-shaped supports (see Fig. 40), sometimes called housings, carrying the guide surface for the slipper. When made of cast steel, they sometimes have the same shape as the cast-iron housings, but are

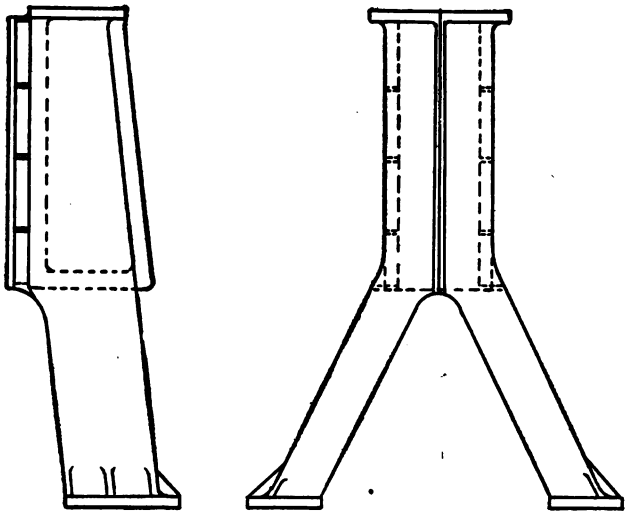


FIG. 41.

made in two parts joined together by a vertical flange (see Fig. 41). The other form in which cast steel is used for engine framing is shown in Fig. 42. These columns are very often used upon the front of the engine, two to each cylinder, in order that access may be had to the crank shaft. When a cross head of the type shown in Fig. 20 is used, four of these columns are used for each cylinder, with a narrow guide surface upon each column. Cylindrical columns of wrought steel are also used, as shown in Fig. 43. When the framing is of

wrought steel it consists of cylindrical columns tied and braced longitudinally and athwartship to make the framing rigid (see Fig. 44).

The framing may consist of housings on both front and

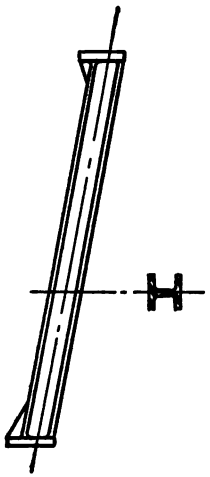


FIG. 42.

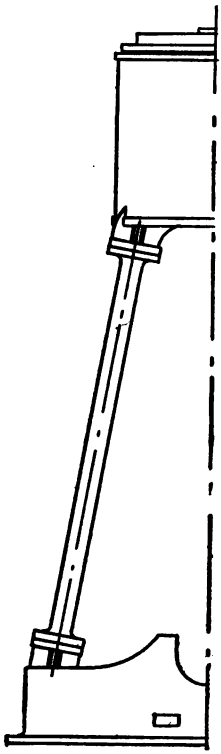


FIG. 43.

back of engine, of housings on the back and columns on the front, of cast steel columns on both front and back of engine, or of steel framing. When housings are used, they should be designed to have sufficient area in the smallest section to carry the maximum load coming upon the piston rod, with a

stress of 600 to 1,000 pounds for cast iron, and 1,000 to 1,500 pounds for cast steel. The shape of the cross section of the housings will be determined by the size of the slipper guide, and the shape of the flange at the top necessary to accommodate the number of bolts needed to carry one-half of the load on the piston rod.

The columns can be figured by formulæ 21 and 22, given for piston rods. The character of the load is alternating, so that a factor of safety of 12 should be used. Since its length will

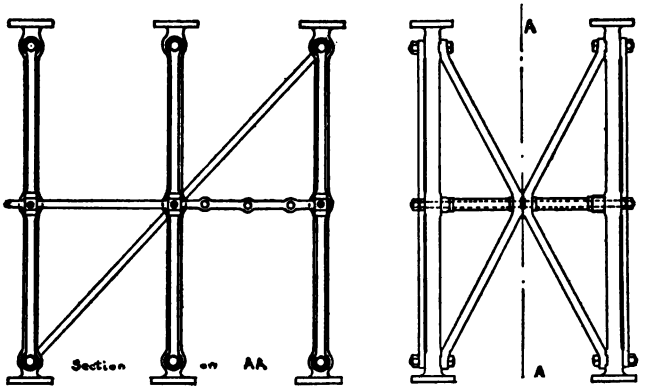


FIG. 44.

be considerably greater than that of the piston rod, and its load only one-half of that, its diameter generally comes about the same as that of the piston rod. The column is very often made hollow, to save weight. Whenever it can be arranged, cross-ties are carried from the top of the housing to the foot of the columns, so as to give the housing additional stiffness. When the framing is made entirely of steel columns and ties the stresses are determined by means of a graphical analysis.

Calculations.—The diameter of the crank shaft and diameter and length of the crank pin have already been determined.

$D = 12$ inches;

$C = 12.5$ inches;

$B = 14'' + 0.25'' = 14\frac{1}{4}$ inches.

The crank shaft will be made built up, in three interchangeable sections.

$A = 0.65 \times 12 = 7.8$ inches (use 8 inches);

$E = 0.28 \times 12 = 3.36$ inches (use $3\frac{1}{4}$ inches);

$F = 0.25$ inches;

$J = 2$ inches;

$K = 12.75$ inches;

$L = 13$ inches;

$M = 11.375$ inches;

$N = 12$ inches.

Assume radius of pitch circle of coupling bolts $= 12'' \times 0.75 = 9'' = r$.

Assume n to be 6.

Formula (43): $$R = \frac{12}{2}\sqrt{\frac{12}{6 \times 9}} = 2.83 \text{ inches.}$$

Let the taper of the bolt be 1 inch per foot, and the length of bolt $\doteq 6\frac{1}{2}$ inches, the large end projecting $\frac{1}{4}$ inch beyond the flange.

Diameter of bolt at large end

$$= 2.83 + \frac{3.5}{12} = 3.122, \text{ (use } 3\tfrac{1}{8} \text{ inches).}$$

The smaller end of the taper will be

$$3.125'' - \frac{6.5}{12} = 2.583 \text{ inches diameter.}$$

The threaded part on the end can be made $2\frac{1}{4}$ inches diameter.

A $2\frac{1}{4}$-inch nut is about 4 inches across the angles, so the pitch circle of the coupling bolts must be at least 2 inches

from the outside of the shaft; 9 inches — (6 inches + 2 inches) = 1 inch clearance between nut and shaft. This can be made less if necessary. Try $r = 8.625$ (formula 42).

$$R = \frac{12}{2}\sqrt{\frac{12}{6 \times 8.625}} = 2.89 \text{ inches;}$$

$$2.89 + \frac{3.5}{12} = 3.182'', \text{ (use } 3\tfrac{1}{4} \text{ inches).}$$

$$\text{Diameter at smaller end of taper} = 3.25 - \frac{6.5}{12} = 2.708 \text{ inches.}$$

The diameter of threaded part can be $2\frac{1}{2}$ inches. The nut will be about $4\frac{1}{2}$ inches across angles; 8.625 — (6 + 2.25) = 0.375 inch clearance between nut and shaft. Diameter of flange = 2(8.625 + 3) = 23.25 inches.

Loads Upon Bearings.—Values of f from Fig. 31 for 3,000 *I. H. P.*, built-up shaft, give the following loads upon the main bearings:

First cylinder, forward

$$L = \frac{21{,}000}{850} \times 0.77 \times 1{,}000 = 19{,}000 \text{ pounds}$$

First cylinder, aft

$$L = \frac{21{,}000}{850} \times 0.77 \times 1{,}000 = 19{,}000 \text{ pounds.}$$

Second cylinder, forward

$$L = \frac{21{,}000}{850}(1{,}000 + 0.11 \times 1{,}000) = 27{,}400 \text{ pounds.}$$

Second cylinder, aft

$$L = \frac{21{,}000}{850}(1{,}000 + 0.61 \times 1{,}000) = 39{,}800 \text{ pounds.}$$

Third cylinder, forward

$$L = \frac{21{,}000}{850}(2{,}000 - 0.14 \times 1{,}000) = 46{,}000 \text{ pounds.}$$

Third cylinder, aft

$$L = \frac{21{,}000}{850}(2{,}000 + 0.64 \times 1{,}000) = 65{,}200 \text{ pounds.}$$

Area for each high-pressure bearing

$$= \frac{19{,}000}{200} = 95 \text{ square inches,} \qquad \frac{95}{12} = 7.92''$$

Area for second cylinder, forward — use 9 inches.

$$= \frac{27{,}400}{250} = 110 \text{ square inches} \qquad \frac{110}{12} = 9.16''$$

Area for second cylinder, aft

$$= \frac{39{,}800}{275} = 145 \text{ square inches} \qquad \frac{145}{12} = 12.1''$$

Area for third cylinder, forward — use 12½ inches.

$$= \frac{46{,}000}{300} = 153.5 \text{ square inches} \qquad \frac{153.5}{12} = 12.8''$$

Area for third cylinder, aft

$$= \frac{65{,}200}{350} = 187 \text{ square inches} \qquad \frac{187}{12} = 15.58''$$ — use 15½ inches.

See Fig. 46 for sketch of the three sections of shafting.

Loads for which main bearing caps and bolts are to be figured are:

High-pressure bearings, $19{,}000 \times 2 = 38{,}000$
Medium-pressure forward bearings, $27{,}400 \times 1.75 = 47{,}900$ — use 48,000 pounds.

Medium-pressure after bearings,

Low-pressure forward bearing,

$$\left.\begin{aligned} 39{,}800 \times 1.75 &= 69{,}700 \\ 46{,}000 \times 1.67 &= 76{,}800 \end{aligned}\right\} \text{use } 75{,}000 \text{ pounds}$$

Low-pressure after bearing,

$$65{,}200 \times 1.4 = 91{,}250 \text{ pounds.}$$

Main bearing bolts for 48,000-pound load must be two of $2\frac{3}{4}$ inches.
Main bearing bolts for 75,000-pound load must be two of $3\frac{1}{4}$ inches.
Main bearing bolts for 91,250-pound load must be two of $3\frac{1}{2}$ inches.

Make lower brass "half-round" with a thickness of $\frac{D}{10} +$ 0.25 inch $= 1.2$ inches $+ .25$ inch $= 1.45$ inches (use $1\frac{1}{2}$ inches). Bolts will clear shaft by $2\frac{1}{2}$ inches to allow for the brass and the metal of the bed.

Distance between centers of $2\frac{3}{4}$-inch bolts

$$= 12'' + 5'' + 2.75'' = 19.75 \text{ inches.}$$

Distance between centers of $3\frac{1}{4}$-inch bolts

$$= 12'' + 5'' + 3.25'' = 20.25 \text{ inches.}$$

Distance between centers of $3\frac{1}{2}$-inch bolts

$$= 12'' + 5'' + 3.5'' = 20.5 \text{ inches.}$$

Handholes in main bearing caps to be $2\frac{1}{2}$ inches wide.

Caps to be of cast steel, and stress to be $\frac{60{,}000}{10} = 6{,}000$ pounds.

Caps for high-pressure and forward medium-pressure bearings:

Formula (45):

$$H = \sqrt{\frac{48{,}000 \times 19.75}{5.5 \times 6{,}000}} = 5.36 \text{ inches.}$$

Caps for after medium-pressure and forward low-pressure bearings:

$$H = \sqrt{\frac{75{,}000 \times 20.25}{8.5 \times 6{,}000}} = 5.47 \text{ inches.}$$

Caps for after low-pressure bearing:

$$H = \sqrt{\frac{91{,}250 \times 20.5}{10.5 \times 6{,}000}} = 5.45 \text{ inches.}$$

Make all of the caps 5½ inches thick from the back of the white metal, or 6 inches thick, allowing for the white metal.

STEAM SPEEDS AND VALVE DIAGRAM.

The steam speeds commonly given for engines are based upon the piston displacement. The volume of steam entering a cylinder is assumed to be the volume given by the product of the piston speed and the area of the cylinder. Steam speed

$$= \frac{A \times PS}{a},$$

where A is the area of the piston in square inches, and a is the area of the passage in the same units. The width of the port in the direction of motion of the valve should be from 0.6 to 0.8 of the eccentricity.

The eccentricity is usually given in inches, half inches, or quarter inches, so it is well to start the solution of the valve gear problem by assuming an eccentricity appropriate to the size of the engine.

I. H. P. of engine	500 to 1,000	1,000 to 2,000	2,000 to 5,000	5,000 to 10,000
Eccentricity:				
H. P. and M. P.....	3″ to 3½″	3½″ to 4″	4″ to 4½″	4½″ to 5″
L. P.....	3″ to 3¾″	3½″ to 4¼″	4″ to 5″	4½″ to 5¼″

In order that the low-pressure valve may not be too broad, when a slide valve is used, it is sometimes necessary to make the valve double ported, and to increase the eccentricity. The breadth of the passage, going from the face of the slide valve

to the cylinder, must be not more than 0.85 D to 0.95 D, where D is the diameter of the cylinder. The port area necessary will depend upon the steam speeds, and the steam speeds must be selected with reference to the desired economy. The steam speeds should be about as follows for engines whose design factor is 0.7 or more:

Main steam pipe.....	6,000 to 7,200 feet per minute.
Throttle valve.....	5,000 to 6,000 feet per minute.

	High-pressure Ports.	Medium-pressure Ports.	Low-pressure Ports.
Entering steam.....	5,000 to 6,000	6,000 to 7,000	7,000 to 8,500
Exhaust steam.....	4,000 to 5,000	5,000 to 6,000	6,000 to 7,000
Exhaust to condenser.			6,000 to 6,500

Engines whose design factor lies between 0.7 and 0.6 can have the following steam speeds:

Main steam pipe.....	6,500 to 7,500 feet per minute.
Throttle valve.....	5,500 to 6,500 feet per minute.

	High-pressure Ports.	Medium-pressure Ports.	Low-pressure Ports.
Entering steam.....	6,000 to 8,000	7,500 to 10,000	9,000 to 13,000
Exhaust steam.....	5,000 to 6,000	6,500 to 8,000	8,000 to 10,000
Exhaust to condenser			7,000 to 9,000

It is best to select the speed of exhaust steam, and from this get the necessary area of ports by means of the formula:

$$a = \frac{A \times PS}{\text{speed of exhaust}}, \qquad (47)$$

where A = area of cylinder;

PS = piston speed.

The breadth of the port having been taken from 0.85 D to 0.95 D, and allowance having been made for the ribs, the width of port can be found. If this is from 0.65 E to 0.8 E,

where E is the eccentricity selected, the valve can be single ported; if more than that, it will be best to make it double ported, and to decrease the breadth of port so that each port shall be from 0.65 E to 0.8 E. The speed of the entering steam cannot be selected arbitrarily, but will be determined by the maximum port opening obtained from the valve diagram.

Entering speed steam: exhaust speed:: maximum port opening: width of port.

If the speed of the entering steam does not come within the limits given, the eccentricity or the width of port should be changed.

The area of the pipe, if one is used, or of each pipe, if two are used, going from one cylinder to the next, should be intermediate between the area of the exhaust ports in one cylinder and the area of the ports at maximum port opening in the next cylinder.

When piston valves are used, the breadth of the steam passage does not have to be considered, as the steam is admitted around the circumference of the valve, and the passage can be so designed as to take care of any reasonable diameter of valve or valves. In the slide valve, allowance has to be made for two or three ribs in the passageway; so, in the piston valve, the entire circumference of the valve is not clear opening. Bridges have to be carried across, to prevent the rings of the piston valve from springing out into the passageway, and these bridges usually take up about a quarter of the circumference. Allowing for this, the diameter of the piston valve can be obtained from the formula:

$$d = \frac{D^2 \times PS \times c}{V \times W}, \tag{48}$$

where D = the diameter of the cylinder in inches;

PS = the piston speed of the engine in feet per minute;

V = the velocity of the exhaust in feet per minute;

W = the width of the port in inches;
c = 0.333 for 0.75 of the circumference clear opening;
0.357 for 0.7 of the circumference clear opening;
and 0.312 for 0.8 of the circumference clear opening.

The lead upon one end of the valve must be assumed before the valve diagram can be started. The lead upon the top end of the cylinder is generally made about $\frac{E}{6}$, and the difference in the cut-offs is so chosen that the lead upon the bottom shall be $\frac{1}{8}$ inch more. Both leads cannot be assumed if the cut-off positions are assumed; one lead can be assumed and the other worked out.

The eccentricity gives us the radius of the circle *ABCD*, Fig. 45. After drawing the two axes *AB* and *CD* at right angles, an arc can be struck from *A* with a radius equal to the assumed lead at the top. The cut-offs which were calculated for the cylinders at the very first are the mean cut-offs of the two ends of the cylinders. Equality of cut-off in percent of the stroke is never attempted in marine engines, where the ratio of connecting rod, length to length of crank, is so small that an impractical inequality of lead would result.

The cut-off on the down stroke is usually 7 or 8 percent greater than the cut-off on the up stroke. The excess of lead upon the bottom should be sufficient to allow for the greater forces to be overcome, and for the fact that, as wear takes place in the joints of the valve gear, the valve is going to drop down, and cause the lead upon the bottom to decrease.

The desired difference in lead can be obtained by laying off from the top end of the diagram a percentage of the diameter of the valve circle *AF*, equal to the desired mean percentage of the cut-off. A perpendicular dropped from the point *F* upon the valve circle will give a point through which a diameter *GH* can be drawn, that will give the cut-off position of the cranks upon the down and up strokes, which will be

accompanied by equal leads. The desired inequality of lead, ⅛ inch, can be obtained by laying back 1/16 inch from *G* to *J*, and forward 1/16 inch from *H* to *K*, for the position of cut-off to give unequal leads. Draw from *J* a tangent to the lead arc at *A*. Bisect the line *JL* by a diameter *MN* of the circle. Upon this diameter draw the two eccentric circles *MO* and

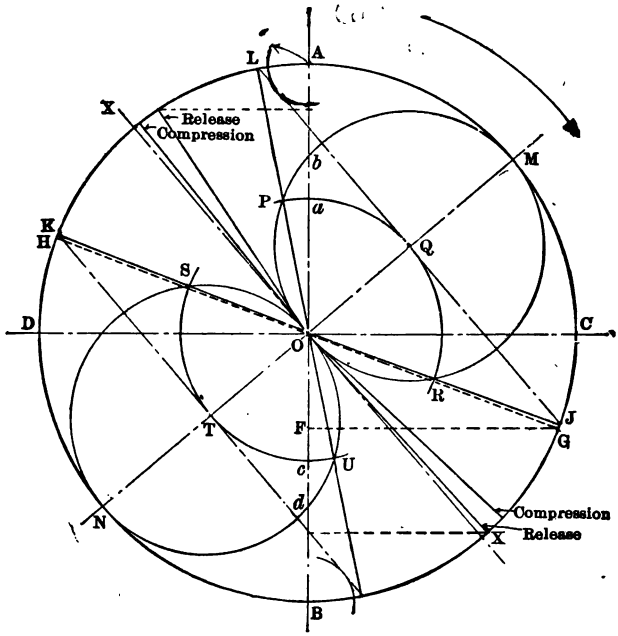


FIG. 45.

NO. Draw the lap circles *PQR* and *STU*. The angle *MOC* is the angle of advance, *MQ* is the maximum port opening at the top of the cylinder, and *TN* at the bottom. *OQ* is the steam lap for the top end of the valve, and *OT* for the bottom end.

No fixed rule can be given for finding the exhaust laps for the valve. Usually the release occurs from 7 to 15 percent

before the end of the stroke, being earlier in fast running engines. The release upon the two ends can be made to occur at the same percent before the end of the stroke, but it is better to have the release upon the up stroke occur a little later than that upon the down stroke, as the dropping of the valve, due to wear, will cause it to occur gradually earlier upon the up stroke and later upon the down stroke. The compression due to this exhaust lap must be considered also, and if it occurs too early in the stroke, so that the engine would run badly, the release must be put earlier, to give a reasonable compression. This is especially important if the cut-off is somewhat short, in which case a rather large negative exhaust lap is required upon one end.

The best method of determining the amount of exhaust lap, and whether it is positive or negative, is to draw the line *XX* perpendicular to *MN*, and note the position of the release points relative to this line, which gives the position of the crank for the midposition of the valve. If the release on the down stroke occurs before the crank reaches *XX*, the exhaust lap is negative, if after the midposition it is positive.

The amount of lap will be the perpendicular distance between the release position and *XX*. Usually the exhaust lap upon the top is from —⅛ inch to —⅜ inch, and upon the bottom from ½ inch to —⅛ inch. There is no harm done if both laps are negative, thus permitting the two ends of the cylinder to be in communication at times, and no effect upon the indicator cards will be noticed until the sum of the exhaust lap amounts to about —½ inch. The distances *ab* and *cd* are also equal to the leads. It is well to tabulate the results, as shown later on in Table VIII.

If steam is taken at the middle rather than at the ends of the valve, the calculations are no different, the angle of advance of the eccentrics is increased by 180 degrees when they are placed upon the shafts, and the steam laps are placed

nearer to the middle of the valve, instead of upon the ends.

The maximum port opening having been determined, and the width of port having been adjusted to give reasonable steam speeds, the final breadth of port in the case of the slide valve, or diameter of valve in the case of the piston valve, can be determined.

When piston valves are used, the high-pressure cylinder has one valve, the medium-pressure one or two, depending upon the size of the valves, and the low-pressure two or four valves. Two smaller valves are used in place of one valve when the one is more than half the diameter of the cylinder, or when the engine can be shortened materially by putting two smaller valves on either side of the center line, rather than one large one on the center line. When two valves are substituted for one, the diameter of each is half the diameter of the one for which they are substituted, as the steam is admitted around the circumference of the valves, and the circumference varies as the diameter.

The valves must be so located that their covers will not foul each other or the cylinder covers, and their location must also suit the crank shaft, in order that the ahead eccentric may be placed in the plane of the center line of the valve or valves. The distance between the center lines of the valves, when two valves are used for one cylinder, should be not less than $y = 1.6\ V + 1$ inch, nor more than $y = 1.8\ V + 2\frac{1}{2}$ inches, where $V =$ the diameter of the piston valves. The distance between the center line of the cylinder and the center line of the valves can be calculated, when the distance between valve centers has been selected, by letting this latter distance be one side of a right triangle, of which the hypotenuse is the sum of the radii of the cylinder cover and valve chest cover plus the desired clearance between the edges of the covers. This clearance varies from nothing to 6 inches, being determined sometimes by the arrangement of crank shaft.

$$X = \sqrt{\left(\frac{b}{2} + \frac{c}{2} + d\right)^2 - a^2}, \qquad (49)$$

where X = distance between center line of valves and cylinder, measured parallel to the center line of the engine;

b = diameter of cylinder cover;

c = diameter of valve chest cover;

d = clearance between covers;

and a = distance between center lines of valves.

The distance d should be sufficient to allow the steam passage between the barrel of the cylinder and the wall of the valve chest to be of the required area, without the passage being made too broad.

The accurate location of the center lines of the cylinders involves the finding of the diameters of the outside of cylinder covers and valve chest covers. Referring back to where the size of bolts used in the cylinder covers was determined, formula (18) gives the diameter of the inner surface of the barrel where the cylinder cover fits, and the diameter of the pitch circle of the bolts is taken as $C + 3d$, where C is the diameter referred to above, and d is the diameter of the studs used in the cover. The flange should extend beyond the pitch circle, an amount equal to $1\frac{1}{2}\ d$, so that the diameter of the outside of the cylinder cover should be:

$$E = D + 2L + 2J + 6d, \qquad (50)$$

where E = the diameter of the outside of the cylinder cover,

D = the diameter of the cylinder;

L = the thickness of the liner;

J = the thickness of the jacket space;

and d = the diameter of the cover stud.

The cover for the piston valve chest must be large enough to permit the liner to be put in. The diameter A having been determined by formula (48), and the thickness of the valve liner T being known, the diameter of the opening under the

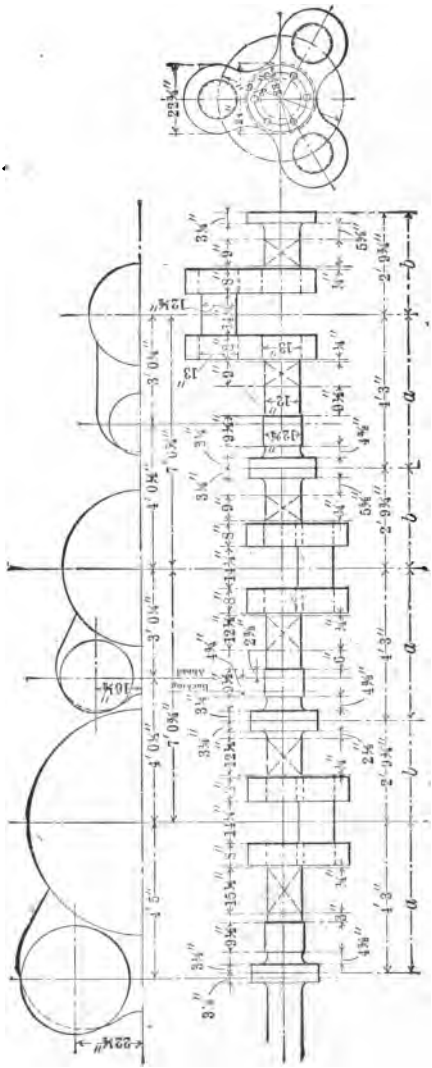


FIG. 46.

cover will be $A + 2T + \frac{1}{2}$ inch. The cover studs are usually 1 inch or $1\frac{1}{8}$ inches in diameter, with the same spacing as recommended for the cylinder cover studs, and the joint is usually $3d$ broad. The outside of the valve chest cover will then be:

$$F = A + 2 \times T + S \qquad (51)$$

where F = the outside diameter of the valve chest cover;

A = the diameter of the valve;

T = the thickness of the liner;

and $S = 6\frac{1}{2}$ inches or $7\frac{1}{4}$ inches, depending upon whether 1-inch or $1\frac{1}{8}$-inch studs are used.

With these diameters, and the details and clearances of the shafting, a sketch like Fig. 46 can be made, locating the center lines of the cylinders and valves. The question as to whether or not the sections of the crank shaft are to be interchangeable must be decided before the distance between the center lines of the cylinders can be determined. If the sections are to be interchangeable, then the dimensions a and b (Fig. 46) must be the same on all sections, and the distances between the cylinder centers will be all the same, and equal to $a + b$. If the eccentrics for the low-pressure cylinder are to be on the low-pressure section of the crank shaft, then the minimum values of a and b will be determined by the low-pressure cylinder. If the eccentrics are to be on the thrust shaft or on the coupling, as shown in Fig. 46, then the minimum value of a will be determined by the medium-pressure cylinder, care being taken that a is not made so small that there is not room for the eccentric pad which must be provided for upon the low-pressure section, even though it is not used for that cylinder. Sometimes all of the eccentrics are placed upon the couplings.

When the crank shaft sections are not made interchangeable, the engine can generally be made shorter. A good arrangement of cylinders and valves is to have the medium-pressure

and low-pressure valves grouped together between those two cylinders, and the high-pressure valves between the high-pressure and medium-pressure cylinders. This makes the piping very compact, although the over-all length of the engine may not be any less than with the other arrangement. In a four-cylinder engine all of the sections are not usually interchangeable; but the first two sections are interchangeable, and so are the last two sections.

Fig. 47 gives arrangement data for several engines. Such a table is convenient for checking calculated results, and for estimating the distance between cylinder and valve center lines when there is not time for calculations.

It is very often convenient to know the length, breadth and height of an engine approximately. It will be found that the length of the engine over-all is from 1.8 ΣD to 1.9 ΣD, when the sections of shafting are not made interchangeable, and from 1.9 ΣD to 2 ΣD when the sections are interchangeable. ΣD is the sum of the cylinder diameters. The height of the engine from the bottom of the bed to the top of the covers is from 5 to 5¼ times the stroke for naval engines, and from 5½ to 6 times the stroke for merchant engines. The breadth of the bed varies considerably, but is almost twice the low-pressure diameter in merchant engines, and from 1½ to 2 times the low-pressure diameter in naval engines, depending upon the type of framing.

Calculations.—Make all valves piston valves. Assume eccentricity = 4¼ inches; lead upon top end = ⅝ inch for high-pressure and medium-pressure and ¾ inch for low-pressure.

As the design factor used in the first part of the calculations was 0.7, the following steam and exhaust speeds in feet per minute will be used:

Cylinder	High	Medium	Low
Entering steam.....	6,000	7,500	9,000
Exhaust steam.....	5,000	6,000	7,000

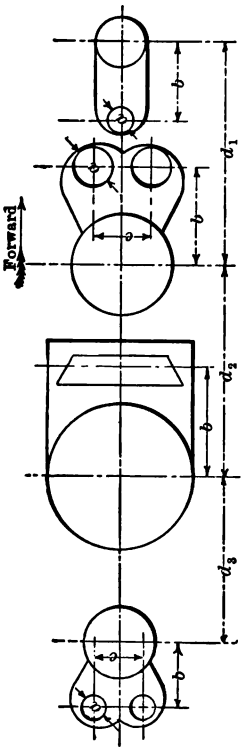


FIG. 47.—ARRANGEMENT KNOWN AS HAP-COD-LOS-MAD.

H—High-pressure cylinder.
M—First intermediate cylinder.
C—Second intermediate cylinder.
L—Low-pressure cylinder.
O—Valve is placed forward of cylinder.

A—Valve is placed aft of cylinder.
S—Cylinder has slide valve.
P—Cylinder has piston valve.
D—Cylinder has twin piston valves.
Y—Cylinder has four piston valves.

Speed of steam through throttle, 5,500 feet per minute.

Speed of steam in exhaust to condenser = 6,500 feet per minute.

The cut-offs obtained in the beginning were *H. P.* = 0.675, *M. P.* = 0.60, *L. P.* = 0.65. When the maximum port opening has been determined, the width of the port can be chosen to give these speeds. The means of the maximum port openings obtained from the diagram, of which only that for the high-pressure is shown (see Fig. 45), are as follows:

Number.	I. H. P.	R. P. M.	Boiler Pressure.	Cylinder Diameters.	Stroke.	Sequence of Cylinders, Type and Location of Valves.	First Cylinder.	
							a.	*b.*
41	600	220	200	8 –13½–22	18	HOP–MOP–LOD	4½″	14″
42	1,100	140	...	17 –25 –43	30	HOP–MOP–LOS	8½″	22″
43	1,500	80	175	22 –36 –59	42	HOP–MOP–LOS	10″	2′ 6″
44	1,500	85	200	19 –31 –54	42	HAP–MOD–LOD	8″	2′ 5″
45	2,250	100	160	23½–39 –65	42	HOP–MOS–LOS	10″	2′ 6½″
46	2,800	75	190	27 –44 –74	51	HOP–MOP–LAD	13″	2′ 11½″
47	3,750	100	165	28 –46 –76	48	HAP–MOD–LOD	16″	3′ 0½″
48	.....	...	...	19–28½–41–60	42	HAP–MOD–LOD–COD	8″	2′ 8″
49	4,000	100	165	32–52–(2)60	42	HAP–MOD–LOD–LAD	18″	3′ 3½″
50	.....	100	215	23–33½–48½–70	51	HAP–MOD–COD–LOS	15″	2′ 1″
51	5,000	85	170	32½–54 –89½	54	HAP–MOD–LOD	22″	3′ 9″
52	7,500	120	200	38½–59 –92	42	HAP–MOD–LY	16″	3′ 7″
53	8,000	86	200	35–50–70–100	66	HOP–LY–CAD–MAD	20″	3′ 9″
54	10,000	85	175	42–66½–(2)77	60	HAP–MAD–LAD–LAD	30″	4′ 10″

Number.	d_1.	Second Cylinder.			d_2.	Third Cylinder.			d_3.	Fourth Cylinder.		
		a.	*b.*	*c.*		*a.*	*b.*	*c.*		*a.*	*b.*	*c.*
41	2′ 4½″	6½″	15½″	.....	2′ 11″	7½″	18½″	13″	.....	.....	.....	.....
42	3′ 9½″	9½″	23½″	.....	4′ 10″	.....	2′ 10″	.....	.....	.....	.....	.....
43	6′ 4½″	17″	4″	.....	6′ 4½″	.....	3′ 8″	.....	.....	.....	.....	.....
44	6′ 10″	8″	2′ 5″	17″	6′ 10″	16½″	4′ 1″	26″	.....	.....	.....	.....
45	5′ 7½″	.....	3′ 0½″	.....	7′ 4½″	.....	4′ 4″	.....	.....	.....	.....	.....
46	7′ 1″	25½″	3′ 9½″	.....	7′ 1″	25½″	4′ 7″	3′ 6″	.....	.....	.....	.....
47	8′ 0½″	18½″	3′ 2½″	2′ 6″	8′ 7″	33½″	4′ 11″	4′	.....	.....	.....	.....
48	7′	15″	2′ 9½″	.....	7′	19″	4′ 2″	2′ 4″	7′	12″	4′ 1½″	21″
49	.....	18″	3′ 5″	2′ 6″	.....	18″	3′ 8½″	2′ 6″	.....	18″	3′ 8½″	2′ 6″
50	8′ 4″	15″	2′ 11″	2′ 6″	7′ 7″	23″	3′ 7″	3′	8′ 4″	...	4′ 9″	.....
51	10′ 2″	22″	3′ 9″	3′	10′ 2″	34″	5′ 11½″	4′ 6″	.....	...	.....	.....
52	10′ 7″	20″	3′ 11″	3′ 1″	10′ 9″	20″	5′ 7″	2′ 8″	.....	...	.....	.....
53	9′ 7″	25″	5′ 9½″	5′	10′ 2″	28″	4′ 10½″	4′ 6″	9′ 3″	17″	4′ 5″	2′ 9″
54	10′	30″	4′ 10″	4′ 9″	10′ 4″	36″	5′ 2″	5′	10′ 4″	36″	5′ 2″	5′

High-pressure, $\frac{2.12'' + 2.24''}{2} = 2.18$ inches.

Medium-pressure, $\frac{1.84'' + 1.96''}{2} = 1.9$ inches.

Low-pressure, $\frac{2.25'' + 2.39''}{2} = 2.32$ inches.

When the diagram for the low-pressure cylinder was drawn, it was found that very large valves would have to be used if the eccentricity was to be 4¼ inches, so the eccentricity for that cylinder was made 4¾ inches.

Width of port for high-pressure,

$$= 2.18'' \times \frac{6{,}000}{5{,}000} = 2.61'' \text{ (use } 2\tfrac{5}{8} \text{ inches).}$$

Width of port for medium-pressure,

$$= 1.9'' \times \frac{7{,}500}{6{,}000} = 2.375'' \text{ (use } 2\tfrac{3}{8} \text{ inches).}$$

Width of port for low-pressure,

$$= 2.32'' \times \frac{9{,}000}{7{,}000} = 2.98'' \text{ (use 3 inches).}$$

High-pressure piston valve,

$$= \frac{(23.5)^2 \times 850 \times 0.333}{5{,}000 \times 2.625} = 11.9'' \text{ (use 12 inches).}$$

Medium-pressure piston valve,

$$= \frac{(41)^2 \times 850 \times 0.333}{6{,}000 \times 2.375} = 33.5'' \text{ (use two of 17 inches).}$$

Low-pressure piston valve,

$$= \frac{(64)^2 \times 850 \times 0.333}{7{,}000 \times 3} = 55.2'' \text{ (use two of } 27\tfrac{1}{2} \text{ inches).}$$

The port areas for cylinders and areas for exhaust steam through valves are:

High-pressure $= \pi \times 12'' \times 0.75 \times 2.625'' =$ 74 square inches.
Medium-pressure $= \pi \times 34'' \times 0.75 \times 2.5'' =$ 200 square inches.
Low-pressure $= \pi \times 55'' \times 0.75 \times 3'' =$ 389 square inches.

Areas for entering steam are:

High-pressure, $74 \times \frac{2.18}{2.625} = 61.5$ square inches.

Medium-pressure, $200 \times \frac{2.02}{2.5} = 162$ square inches.

Low-pressure, $389 \times \frac{2.21}{2.875} = 299$ square inches.

Throttle valve and main steam pipe:

$$\frac{\pi(23.5)^2 \times 850}{4 \times 5{,}500} = 66 \text{ square inches. Use } 9\tfrac{1}{4}\text{-inch pipe.}$$

Pipe or pipes between high-pressure and medium-pressure valves:

$$\frac{74 + 162}{2} = 118 \text{ square inches. Use } 12\tfrac{1}{4}\text{-inch pipe.}$$

Pipe or pipes between medium-pressure and low-pressure valves:

$$\frac{200 + 299}{2} = 249.5 \text{ square inches. Use } 17\tfrac{3}{4}\text{-inch pipe.}$$

Pipe to condenser from low-pressure cylinder:

$$\frac{\pi(64)^2 \times 850}{4 \times 6{,}500} = 421 \text{ square inches. Use 23-inch pipe.}$$

The data for the valve gear should be collected, and given in some such form as shown in Table VIII. The words top and bottom at the heads of the columns refer to the steam upon the top side and upon the under side of the piston; the quantities given for release and compression are those resulting from the exhaust lap in the same column, the release occurring at one end of the cylinder and the compression at the other end.

TABLE VIII.

	High-Pressure.		Medium-Pressure.		Low-Pressure.	
Eccentricity	4¼ inches		4¼ inches		4¼ inches	
Number and diameter of valves	1—12-inch		2—17-inch		2—27½-inch	
Steam Taken at	Middle.		Middle.		Ends.	
	Top.	Bottom.	Top.	Bottom.	Top.	Bottom.
Width of ports	2½″	2½″	2½″	2½″	3″	3″
Steam lap	2⅝″	2″	$2\frac{13}{32}$″	$2\frac{9}{32}$″	2½″	2½″
Exhaust lap	—⅛″	+7/16″	+⅛″	+13/16″	—1/16″	+⅛″
Angle of advance	40° 30′	.....	45° 30′	.....	42° 10′	.....
Steam lead, linear	⅛″	¼″	⅛″	¼″	11/16″	15/16″
Cutoff	0.715	0.635	0.64	0.56	0.69	0.61
Steam release before end of stroke	0.11	0.10	0.11	0.10	0.12	0.11
Steam compression before end of stroke	0.13	0.14	0.13	0.14	0.145	0.155
Maximum port opening	2⅝″	2½″	$1\frac{27}{32}$″	$1\frac{31}{32}$″	2½″	2½″
Maximum exhaust opening	2⅝″	2½″	2⅝″	2⅝″	3″	3″
Velocity of steam	6,180	5,820	7,750	7,250	9,250	8,750
Velocity of exhaust	5,000	5,000	6,000	6,000	7,000	7,000

The extreme diameters of the cylinder covers are as follows:

High-pressure, $23.5'' + 2 \times 1.125'' + 2 \times 0.75'' + 6 \times 1.25'' = 34.75''$

$34.75'' - 23.5'' = 11.25''$

Medium-pressure $41'' + 11.25'' = 52.25''$

Low-pressure, $64'' + 11.25'' = 75.25''$

The extreme diameters of valve chest covers are as follows: High-pressure piston valve = 12 inches inside diameter. Thickness of liner = 1 inch. The diameter of the hole under the cover = 12 inches + 2 × 1 inch + ½ inch = 14½ inches.

Now, $185 \times (14.5)^2 \times 0.7854 = 30{,}500$ pounds. If 1-inch studs are used the pitch circle will be $14\frac{1}{2}$ inches $+ 3$ inches $= 17\frac{1}{2}$ inches diameter.

$\frac{30{,}500}{2{,}060} = 14.8$; use 16 studs. $\quad \frac{17.5'' \times \pi}{16 \times 1''} = 3.44$ diameters apart.

Cylinder.	Valve Diameter.	Diameter of Stud.	Pitch Circle of Studs.	Number of Studs.	Spacing of Studs in Diameters.
H. P.	12″	1″	$17\frac{1}{2}$″	16	3.44
M. P.	17″	1″	$22\frac{1}{2}$″	17	4.15
L. P.	$27\frac{1}{2}$″	1″	33″	20	5.18

By Formula (51):

High-pressure valve chest cover

$$= 12'' + 2 \times 1'' + 6\tfrac{1}{2}'' = 20\tfrac{1}{2} \text{ inches.}$$

$$20\tfrac{1}{2}'' - 12'' = 8\tfrac{1}{2} \text{ inches.}$$

Medium-pressure valve chest cover

$$= 17'' + 8\tfrac{1}{2}'' = 25\tfrac{1}{2} \text{ inches.}$$

Low-pressure valve chest cover

$$= 27\tfrac{1}{2}'' + 8\tfrac{1}{2}'' = 36 \text{ inches.}$$

The distance between the center lines of the medium-pressure valves, $= y$, should not be less than $1.6 \times 17.0'' + 1'' = 28.25$ inches, nor more than $1.8 \times 17.0 + 2.5 = 33$ inches. It was found best to use $y = 33$ inches, in order to keep the engine as short as possible.

X for the medium-pressure valves

$$= \sqrt{\left(\frac{52.25'' + 25\frac{1}{2}''}{2} + 1''\right)^2 - (16\tfrac{1}{2}'')^2} = 36.25'' \text{ nearly.}$$

For the low-pressure valves: $1.6 \times 27.5'' + 1'' = 45$ inches; $1.8 \times 27.5'' + 2.5'' = 52$ inches. Make $y = 45$ inches.

In order that the low-pressure valve cover may clear the covers of the medium-pressure valves by 1 inch, the distance between these valves and the center line of the low-pressure cylinder should be

$$\sqrt{\left[\frac{75.25'' + 25.5''}{2} + 1''\right]^2 - (16\tfrac{1}{2}'')^2} = 48\tfrac{1}{2} \text{ inches, approximately.}$$

The minimum distance between the low-pressure and medium-pressure cylinders will then be 36.25 inches + 48.5 inches = 84.75 inches.

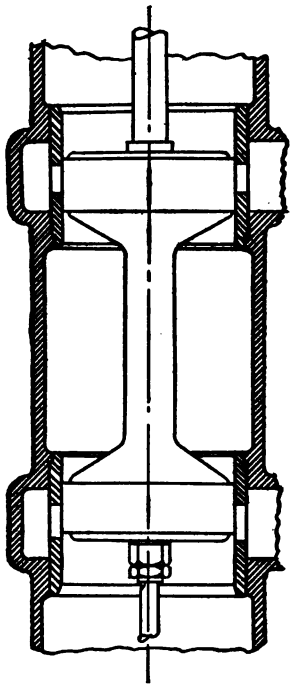


FIG. 48.

VALVES AND VALVE GEAR.

Piston Valves.—Piston valves are made solid, as shown in Fig. 48, or hollow, as shown in Fig. 49. They are made hollow when it is desired to have one pipe supply steam to both ends of the valve, and the area through the middle of the valve should equal the area through the pipe. The length of the

valve depends upon the location of the valve liners, which are placed a sufficient distance from the top and bottom of the valve chest to allow the steam to enter and get away from the ends without the area for steam passage being restricted. The liners should be placed as near the ends as possible, and the passageways to the cylinder made as direct as possible, in

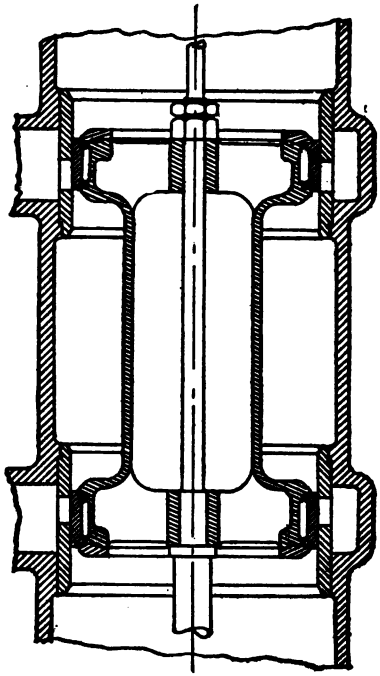


FIG. 49.

order to reduce the clearance space. The length of the liners should be such that the piston valve rings will not spring out at the extremities of the stroke. The liners should be counterbored at the ends sufficiently to allow the rings to over-travel ¼ inch or more. The length of the piston valve liner = width of port + steam lap + exhaust lap + the travel of the valve.

The valve stems, eccentric rods and links must be designed to take care of the frictional load, the inertia and the weight of the valves. In addition, if a single valve is used with a guide, such as shown in Fig. 50, the stem below the valve must be designed for the bending that may come upon it from the pull of the drag rods in reversing when the valve is at its lowest point in the stroke.

In the case of the slide valve, the frictional load can be calculated from the area of the surfaces in contact and the unbalanced pressure upon the back of the valve, no allowance being made for any balancing device. The piston valve is not subjected to any load due to unbalanced pressure, but the friction of the rings and the stuffing-box must be allowed for. It is usual to assume that this load is some multiple of the weight of the valve, valve stem, cross-head and block. If this load is taken as three times the weight of the above parts, a reasonable allowance will be made.

The inertia of the valves is calculated upon the assumption of harmonic motion, and the maximum inertia, at the beginning and end of the stroke, is used. The formula which gives this inertia is:

$$F = 0.00002837WRN^2, \tag{52}$$

where W = weight of valve or valves, valve stem, crosshead and block, usually for the low-pressure gear;

R = length of crank, or eccentricity in inches;

and N = revolutions per minute.

In the case of piston valves, the load for which the valve stem should be figured will be:

$$L = W(4 + 0.00002837RN^2), \tag{53}$$

if no balance piston is used; and

$$L = W(3 + 0.00002837RN^2), \tag{54}$$

if a balance piston is used.

If balance pistons are used which balance more than the weight the load can be still further decreased.

In the case of slide valves the load upon the valve stems will be:

$$L = pAf + W(1 + 0.00002837RN^2), \tag{55}$$

if no balance piston is used; and

$$L = pAf + 0.00002837WRN^2), \tag{56}$$

if a balance piston is used:

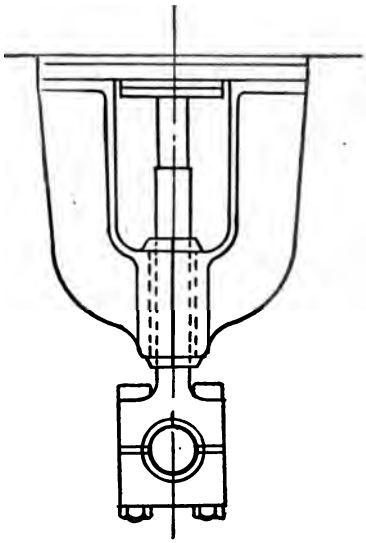


FIG. 50.

where p = the unbalanced unit pressure upon valve;

A = area exposed to unbalanced pressure;

f = coefficient of friction, usually taken as 0.2;

and W, R and N are as above.

The portion of the valve stem between the valve and the link block in the case of single valves, and between the piston valve and the yoke in the case of twin valves should be figured by means of the piston rod formula; the portion of the stem within the valve should be figured for tension only, as the stem

is shouldered down where it enters the valve, and the thrust is carried by this shoulder.

This load L is carried by the links, and when they are at an angle with the horizontal (see Fig. 51) there will be a tendency for the block to slide along the links. As the valve stem is kept from moving by the valve stem guide, this tendency results in a bending moment upon the valve stem, and a reaction in the drag rods to keep the links from moving. The maximum angle that the links make with the horizontal is assumed to be:

$$\sin^{-1} = \frac{2E \sin (90 - d)}{b}, \tag{57}$$

where E = eccentricity of valve;

d = angle of advance;

and b = distance between eccentric rod pins, usually = $6E$.

This formula, upon the assumption of $b = 6E$, becomes

$$\sin^{-1} = \frac{\sin (90^\circ - d)}{3}. \tag{58}$$

The component bending the valve stem will be

$$P = \frac{L \sin (90^\circ - d)}{3}, \tag{59}$$

and the bending moment upon the valve stem will be $M = Pl$, where l is the distance from the bottom of the valve stem guide to the lowest point in the travel of the link block. For twin valves, with the stems yoked together, the bending moment would act upon the yoke, which always has plenty of strength.

If the eccentric rod were normal to the link at the time when the latter makes its maximum angle with the horizontal (see Fig. 51) the load P would be all that the drag rods would have to carry; but since the eccentric rod is in line with the valve stem at that time, a portion of the load normal to the

links will come upon the drag rods, in addition to the component along the links. In the diagram accompanying Fig. 51, AO is the load L acting through the valve stem. This load is resolved into BO along the links and AB normal to the links. Since the eccentric rod is in the position OD, the load $OC =$

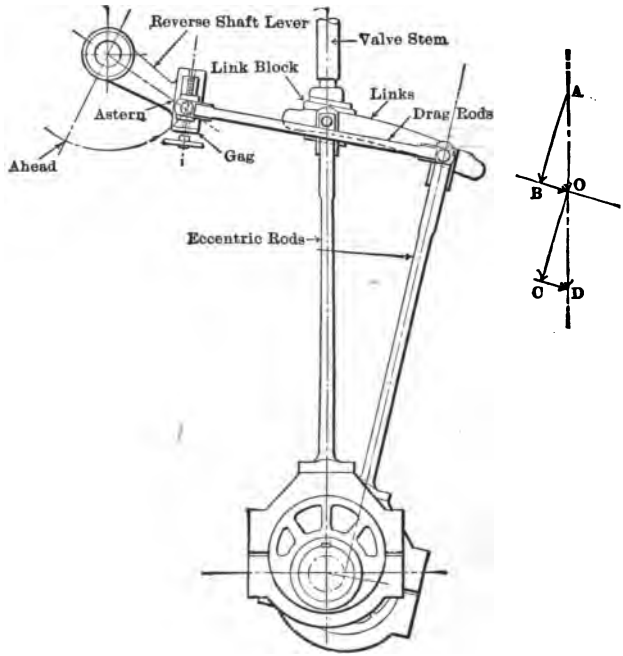


FIG. 51.

AB, normal to the links, is resolved into CD and OD. The drag rods have to take care of $BO + CD = 2P$, and the eccentric rod is subjected to the load L. Although the drag rods are not exactly parallel to the links in the position shown, the load upon the rods will be practically $2P$, so that each rod should be designed for the load P.

The yokes, Fig. 52, should be figured as beams supported at the ends and loaded at the middle with the load L. The surface of the valve stem guide in the case of single valves, and of the valve-yoke guide in the case of twin valves, should be such as to keep the unit bearing pressure, due to the load P, between 70 and 100 pounds.

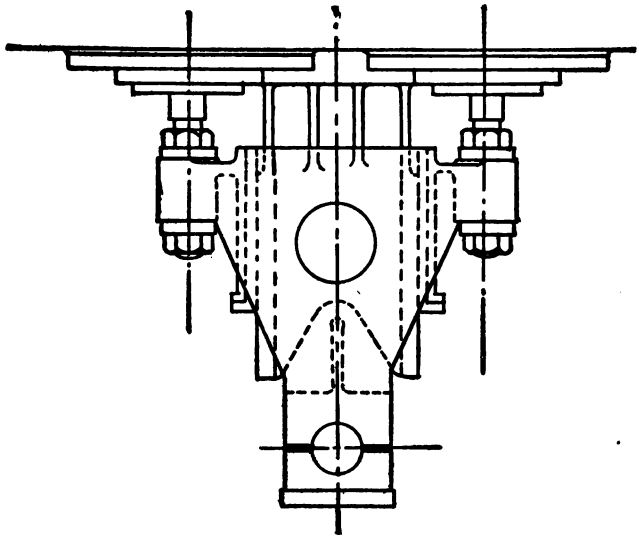


FIG. 52.

The diameter of the eccentric rods and drag rods at the middle should be figured by the connecting rod formula, as they are columns hinged at the ends. The diameter of the drag rods at the ends can be made three-fourths of their diameter at the middle. The diameter of the eccentric rods at the top should be .0.9, and at the bottom 1.1 of the diameter at the middle. The length of the drag rods is usually from $15E$ to $18E$, and of the eccentric rods $20E$ to $30E$ for merchant engines and $15E$ to $20E$ for naval engines. The diameter

of the bolts in the caps, etc., should be such as to carry the loads with the factors of safety given in Table IV.

The link is usually of the double-bar type, Fig. 53, and is figured as a beam supported at the ends and loaded at the middle with the load L; as mentioned before, the length of the beam, or distance between eccentric rod pins, a, is usually $6E$. The breadth of the bars b is usually about one-third of the depth c, and the depth is so chosen that the factor of safety

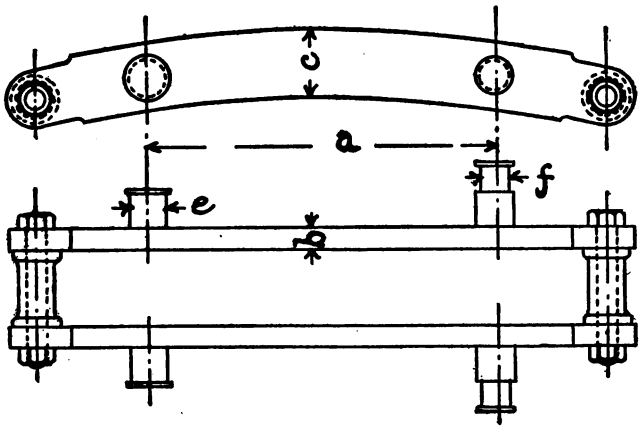


FIG. 53.

will be 12, as the load is alternating. Making this assumption in regard to the relation of breadth and depth, the formula for the depth of each link bar becomes:

$$h = \sqrt[3]{\frac{9La}{4f}}, \qquad (60)$$

where L = load upon valve gear;

a = distance between eccentric rod pins;

and f = allowable stress with factor of safety of 12.

The diameter d of the link block pin, Fig. 54, is from 0.9 to 1.0 of the depth of the bar c; the diameter of the eccentric

rod pins *e*, Fig. 53, is about three-fourths the depth of the link bar, and the diameter of the drag link pin *f* is about three-fourths the diameter of the eccentric rod pin. The lengths of all these pins, as well as that of the block gibs *g*, must be such as to keep the bearing pressures within the limits given in Table VII. The thickness of the metal *h* joining the link block pin at the sides to the sliding gibs should be from 0.25 to 0.3 of the diameter of the link block pin *d*.

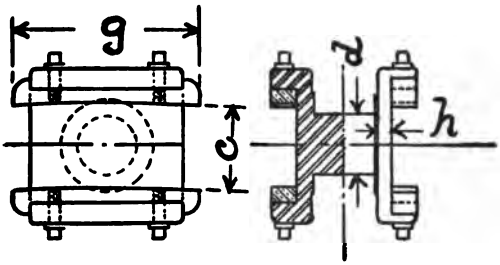


FIG. 54.

The diameter of the eccentric will be:

$$D = 2(r + E + c), \tag{61}$$

where r = radius of eccentric pad on crank shaft;

E = the eccentricity;

$c = \frac{r}{3}$, if the lower part of the eccentric is made of cast iron,

and $= \frac{r}{4}$ if it is made of cast steel.

The upper part of the eccentric, Fig. 55, is always made of cast iron, and joined to the lower part by bolts or collar studs. The keyway should be cut on a line at right angles to the joint of the two parts, so that the eccentric can be readily taken off. If the eccentric is so situated that it can be moved along the shaft clear of the key, then the keyway can be on the side.

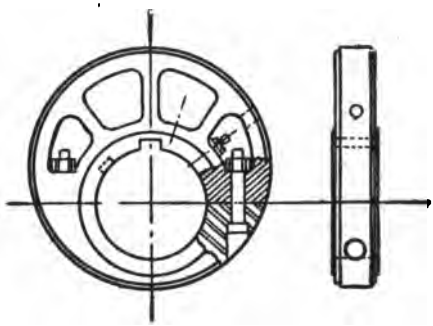


FIG. 55.

as shown dotted in Fig. 55, and the set screw more conveniently located.

It is well to make the keyway considerably broader than the key in the shaft, and to fit liners on either side, so that slight changes can be made in the angular advance, if it is thought best. The breadth of the eccentric should be sufficient to keep the bearing pressure within the limits given in Table VII.

The eccentric strap, Fig. 56, should have lips fitting on each

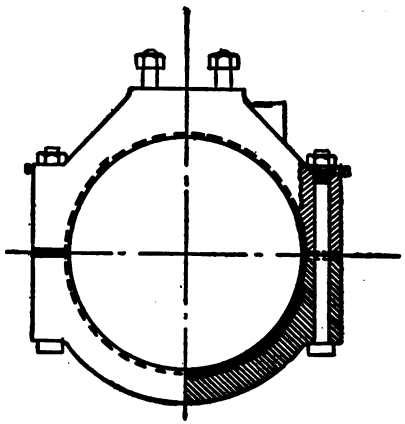


FIG. 56.

side of the bearing surface of the eccentric sheaves, to keep the strap in place. The strap bolts should be designed to carry the load L coming upon the valve gear, as should also the bolts uniting the eccentric rod to the strap. The straps are usually made of cast steel lined with white metal, and the section of the lower half, exclusive of the white metal, should be sufficient to carry the load L with a factor of safety of 8.

The reverse shaft levers, Fig. 51, should be figured for the load $2P$ upon the drag rods, and should be of such a length that the angle moved through is not more than 80 degrees. The "gag" upon the lever should be so arranged that, in the backing position, the center line of the screw is vertical. This will cause the position of the link, when backing, to be practically the same, irrespective of the position of the end of the drag rod in the slot. In the ahead position of the lever the gag screw will be nearly horizontal, and the link can be pulled in an amount about equal to the travel of the nut on the thread.

The reverse shaft should be figured for torsion and bending, due to the thrust of the drag rods. As it is not always possible to get the bearings close to the various reverse shaft levers, the bending moment may be large in these shafts. The equivalent twisting moment should be found by formula 36. The twisting moment T used in this formula should be that coming upon the portion of the reverse shaft nearest the reversing engine. The reversing engine is usually placed near the middle of the length of the main engine, and it is generally safe to figure the shaft for the twisting moment necessary to move the low-pressure gear; for when these links make the greatest angle with the horizontal plane the other links are in a more advantageous position. In the preliminary design the bending moment can be neglected, and the factor of safety increased from 12 to 15 to allow for this neglect. After taking off the medium-pressure reverse shaft levers, the size of the

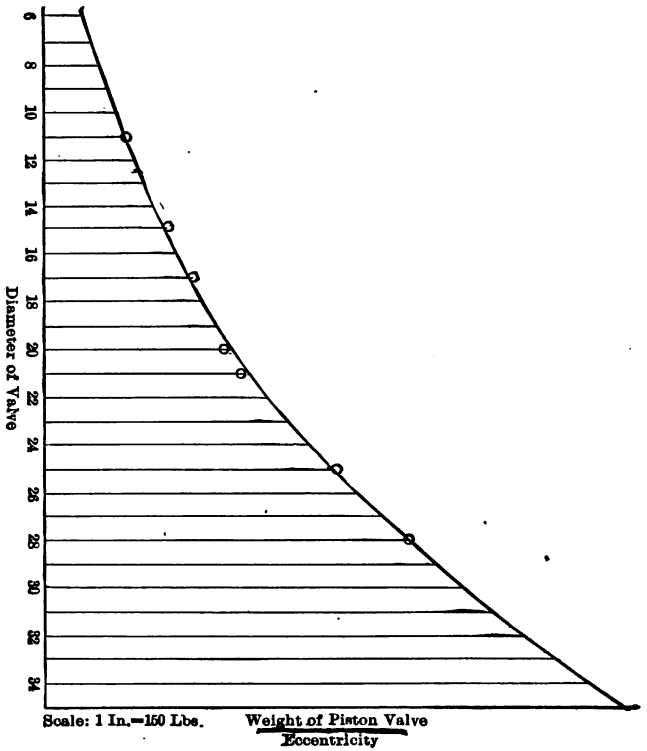


FIG. 57.

shaft running to the high-pressure levers can usually be decreased.

In using formula 52, the weight of the valves, valve stems, cross-heads and blocks is usually obtained from data for similar engines. If no such data are available, Fig. 57 can be used to get the weight of valves for purposes of calculation. The ordinates of the curve are the weights of the valves divided by the eccentricity, so that the weight of any diameter

of valve can be obtained approximately by multiplying the quantity in Fig. 57 by the eccentricity used. The weight of the other parts can be calculated roughly.

Calculations.—Assumed weights of valve gear are as follows:

High-pressure (12 inches diameter):	
1 valve	325 pounds.
1 stem	350 pounds.
1 block	100 pounds.
	775 pounds.
Medium-pressure (two 17 inches diameter):	
2 valves	1,020 pounds.
2 stems	500 pounds.
1 crosshead	325 pounds.
1 block	100 pounds.
	1,945 pounds.
Low-pressure (two 27½ inches diameter):	
2 valves	2,650 pounds.
2 stems	500 pounds.
1 crosshead	400 pounds.
1 block	100 pounds.
	3,650 pounds.

Formula (54):

$$L = [3 + (0.00002837 \times 4.75 \times 120^2)]3{,}650 = 18{,}100 \text{ for low-pressure.}$$

$$= [3 + (0.00002837 \times 4.25 \times 120^2)]1{,}945 = 9{,}100 \text{ for medium-pressure.}$$

Formula (59):

$$P = \frac{18{,}100 \times 0.7431}{3} = 4{,}475 \text{ for low-pressure.}$$

$$P = \frac{9{,}100 \times 0.7}{3} = 2{,}125 \text{ for medium-pressure.}$$

Assume length of drag rods $= 16E = 68$ inches.

Assume length of eccentric rods $= 22E = 94$ inches (approximately).

$$f = \frac{60{,}000}{12} = 5{,}000 \text{ pounds per square inch.}$$

Drag Rods—Low-pressure Gear.

Formula (27):

$$F = \frac{2 \times 4{,}475}{\pi \times 5{,}000} = 0.572$$

$$D^2 = \sqrt{\frac{1.08 \times 0.572 \times 60{,}000 \times 68^2}{10{,}000{,}000}} + 0.327 + 0.572 = 4.76$$

$D = 2.18$ inches (use $2\frac{1}{4}$ inches) at middle of length;

$2.25 \times 0.75 = 1.69$ (use $1\frac{3}{4}$ inches) at ends.

From Table IV., the load, 4,475 pounds, requires two 1-inch bolts to be used in the drag rod caps.

Eccentric Rods—Low-pressure Gear.

$$f = \frac{70{,}000}{12} = 5{,}840;$$

$$F = \frac{2 \times 18{,}100}{\pi \times 5{,}840} = 1.98$$

$$D^2 = \sqrt{\frac{1.08 \times 1.98 \times 70{,}000 \times 94^2}{10{,}000{,}000}} + 3.96 + 1.98 = 13.65$$

$D = 3.69$ inches (use $3\frac{3}{4}$ inches) diameter at middle of rod;

$3.75 \times 0.9 = 3.38$ (use $3\frac{3}{8}$ inches) diameter at top of rod;

$3.75 \times 1.1 = 4.13$ (use 4 inches) diameter at bottom of rod.

The low steam speeds used and the large low-pressure valve resulting, together with the larger eccentricity, make the low-pressure eccentric rods larger than we care to use on the medium-pressure and high-pressure gears, so the rods for the

latter will be calculated from the medium-pressure weights and eccentricity.

With $L = 9{,}100$, $E = 4.25$. Formula 27 gives the diameter at middle of medium-pressure and high-pressure eccentric rods = 3 inches; diameter at upper end of medium-pressure and high-pressure eccentric rods = 2¾ inches; diameter at lower end of medium-pressure and high-pressure rods = 3⅜ inches.

Links.

Formula (60):

$$h = \sqrt[3]{\frac{9 \times 18{,}100 \times 28.5}{4 \times 5{,}840}} = 5.83 \text{ (use } 5\tfrac{3}{4} \text{ inches) for depth of link bar.}$$

$$\text{Breadth of link bar} = \frac{5.75}{3} = 1.92 \text{ inches (use 2 inches).}$$

Diameter of link block pin

$$= 5.75 \times 0.9 = 5.18 \text{ inches (use 5 inches).}$$

Allowable load, Table VII. = 800.

$$\frac{18{,}100}{750 \times 5} = 4.83 \text{ inches (use 5 inches for length of pin).}$$

Length of link block gibs

$$= \frac{18{,}100}{2 \times 2 \times 300} = 15 \text{ inches.}$$

Thickness of metal joining link block pin and gibs

$$= 5'' \times 0.3 = 1.5 \text{ inches.}$$

Diameter of eccentric rod pins

$$= 5 \times 0.75 = 3.75 \text{ inches.}$$

Length of eccentric rod pins

$$= \frac{18{,}100}{2 \times 3.75 \times 750} = 3.22 \text{ inches (use } 3\tfrac{1}{4} \text{ inches).}$$

Diameter of drag rod pins

$$= 3.75 \times 0.75 = 2.82 \text{ inches (use 3 inches).}$$

Length of drag rod pins

$$= \frac{4,475}{3 \times 500} = 2.98 \text{ inches (use 3 inches).}$$

Diameter of low-pressure eccentric, which is on a coupling, (see Figure 46), $= 2(11.625 + 4.75 + 3.25) = 39.25$ inches.

Diameter of medium-pressure and high-pressure eccentrics, $= 2(6.375 + 4.25 + 3.25) = 27.75$ inches.

Width of low-pressure eccentric $= \frac{18,100}{150 \times 39.25} = 3.08$ (use 3 inches). Allowing for ½-inch lip on each side, the total breadth of low-pressure eccentric and strap = 4 inches.

Width of medium-pressure and high-pressure eccentrics $= \frac{9,100}{150 \times 27.75} = 2.19$ (use 3 inches). Allowing for ½-inch lip on each side, the total breadth of high-pressure and medium-pressure eccentric and strap = 4 inches.

Diameter of bolts and studs in eccentric sheaves, see Fig. 56, to be 2 inches, see Table IV.

In order that the reverse shaft levers may not have to move through an angle greater than 80 degrees, the length of the levers should be:

$$\frac{4.75 \times 6}{2 \times \sin 37^\circ 30'} = \frac{14.25}{0.61} = 23.4 \text{ inches (use } 23\tfrac{1}{2} \text{ inches).}$$

We will place the reversing cylinder between the low-pressure and medium-pressure cylinders, and figure the reverse shaft for the twisting moment from the low-pressure gear.

$$T = 2 \times 4{,}475 \times 23.5 = 210{,}000 \text{ inch pounds.}$$

$$f = \frac{70{,}000}{15} = 4{,}670 \text{ pounds.}$$

$$D = \sqrt[3]{\frac{210{,}000 \times 5.1}{4{,}670}} = 6.12 \text{ inches (use 6 inches) for the diameter of the reverse shaft.}$$

Assuming that the turning moment necessary to reverse the high-pressure gear is one-half that for the low-pressure gear, the size of shaft beyond where the medium-pressure reverse shaft levers are taken off can be made 5 inches.

THE DESIGN OF TURNING ENGINES.

When engines are being overhauled, or when the valves are being set, it is often necessary to turn the crank shaft over. In small engines this can be done by means of a bar fitting into holes drilled in one of the coupling flanges. In larger engines, developing from 500 to 1,000 indicated horsepower, a worm and wheel such as is shown in Fig. 1 can be used, and in engines over 1,000 indicated horsepower it is customary to use two worms and two worm wheels connected to a steam engine of one or two cylinders (see Fig. 2). Such an arrangement gives a large multiplication of power in a small space and enables one to use small steam cylinders.

In Fig. 1 the worm is carried in an eccentric bearing, so that by turning the bearing the worm is thrown out of gear and the worm wheel is free to revolve with the crank shaft. In Fig. 2 the large worm *C* is attached to the shaft by means of a removable key or pin, or in some other way, so that when it is not in use it can be screwed to the upper part of the shaft and the worm wheel *D* will be free to revolve. In some cases the shaft of the large worm is not fixed at the lower end, but is held by a rod operated by a hand wheel. The shaft has in the upper part a Hooke's joint, which permits the lower part of the shaft and the worm to be swung back out of gear.

In Fig. 2 the small worm *A* is keyed to the crank shaft of the turning engine, and drives the small worm wheel *B*. Upon the same shaft with the latter is the large worm *C*, engaging with the large worm wheel *D*, which is usually at the aft part of the engine, and when convenient is mounted upon the coupling between the crank shaft and the thrust shaft.

WORM WHEELS.

The turning engine should be of such size as to be able to turn the main engine over once in from five to ten minutes.

The revolutions per minute of the engine are usually between 200 and 400, so that it will make from 1,000 to 4,000 revolutions for one turn of the main engine crank shaft. If the worm *A*, Fig. 2, on the turning engine crank shaft is single-threaded, the turning engine will have to make as many revolutions as there are teeth on the small worm wheel *B* in order to revolve it once. The same is true of the large worm *C* driving the large worm wheel *D* on the main engine crank shaft; hence the number of teeth on the two worm wheels must be such that

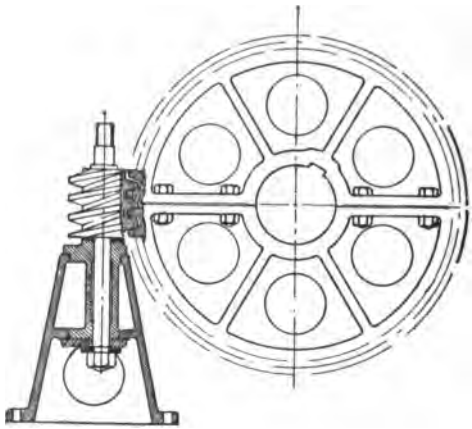


FIG. 1.

$$n\, n_1 = r\, m \qquad (1)$$

n = number of teeth on small worm wheel.
n_1 = number of teeth on large worm wheel.
r = revolutions per minute of turning engine.
m = number of minutes required for one revolution of the main engine crank shaft.

There is no fixed relation between the value of $r\, m$ and the indicated horsepower of the engine. But, roughly speaking, for engines of 1,000 indicated horsepower, $r\, m = 1{,}000$; for

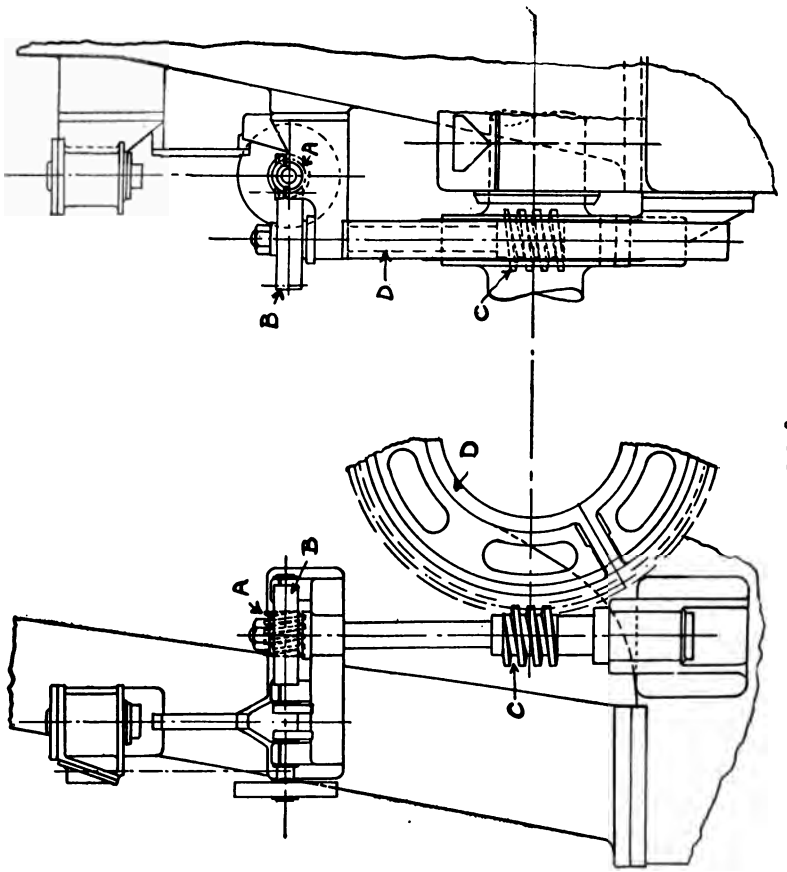


FIG. 2.

engines of 3,000 to 5,000 indicated horsepower, $r\ m = 2{,}000$; for engines of 8,000 to 10,000 indicated horsepower, $r\ m = 3{,}000$.

In order that the large worm wheel may be removed without disturbing anything else, it is usual to make it in two parts, so the number of teeth on this wheel must be even. The diameter of the wheel will be limited by the height of the crank shaft above the foundation. It will be placed as close to a bearing as possible, and it will be found that the diameter of the pitch circle of the wheel cannot be much more than 1.5 times the stroke of the main engine without making it difficult to have access to the holding-down bolts in that neighborhood. The diameter will usually be from 1.1 to 1.5 times the stroke of the main engine.

For practical reasons it is not advisable to use a pitch of less than 1.75 inches upon the small worm wheel, and the pitch is usually from 1.75 to 2.25 inches. The pitch used on the large worm wheel is usually from 2.25 to 3.5 inches.

The proportions of the teeth on the worms and wheels should be about as follows:

Length of teeth = .65 pitch.

Face of teeth = .3 pitch.

Flank of teeth = .35 pitch.

Thickness of teeth at pitch circle = .48 pitch.

Breadth of teeth at root of worm wheels = 2 to 2.5 pitch, or such that the arc of contact between worm and wheel is about 60 degrees.

Least number of teeth on small worm wheel = 25 to 30. When necessary, the thickness of the teeth of the worm wheel at the pitch circle can be made more than .48 pitch if the worm is made of a stronger material than the wheel.

The size of the teeth cannot be determined until the power which they are to transmit is known. The size of the turning engine cylinder, or cylinders, will depend upon the initial

friction of the engine, the efficiency of the transmission gear, and the pressure at which the turning engine takes steam. The turning engine takes steam from the auxiliary steam line, and its pressure may be anywhere from 50 to 100 pounds per square inch.

FRICTIONAL RESISTANCE TO BE OVERCOME.

The power which the turning engine delivers to the crank shaft must be sufficient to start the engine from a state of rest. The frictional resistance to be overcome cannot be taken to be the same as the so-called initial friction determined from engine trials where the engine is run at as low speeds as possible, and the curve of initial friction is continued to the point of zero revolutions. It is well known that for most substances the coefficient of friction is larger in starting from a state of rest than when in motion. The piston speed in these tests for initial friction seldom gets below 200 to 250 feet per minute. When the turning engine is in operation, however, the main engine turns over only once in from five to ten minutes. It is probable, also, that the bearings and piston rods will not be as well oiled at this time as when the engine is under steam, and the friction of the piston rings will be greater, as they have not the lubrication that the condensed steam on the cylinder walls affords. So the initial friction must be multiplied by the ratio that the coefficient of friction from a state of rest bears to the coefficient of friction of motion, and also by a factor to allow for the worst condition of bearings and cylinder walls.

Experiments made by Professor Thurston show that with a cast iron journal in a steel bearing under a pressure of 50 pounds per square inch, and lubricated with lard oil, the coefficient of rest was .07, while when moving at a speed of 150 feet per minute the coefficient was .02. In experiments made by Mr. Wilfred Lewis, reported in the *Proceedings* of the A. S. M. E., vol. 7, p. 273, the coefficient of friction between

the teeth of gears was found to vary, as shown in the following table:

Velocity of sliding feet per minute—

3.15 5.24 10.5 21 52.5 105 210

Coefficient of friction—

.095 .088 .074 .059 .038 .026 .020

When these are plotted, the curve gives the coefficient of

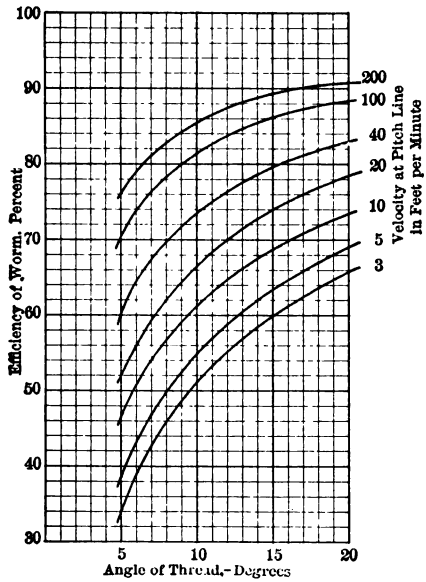


FIG. 3.

friction for a velocity of 0 feet per minute as .108. This makes the coefficient of friction at 0 velocity about 4.5 times that at 150 feet per minute. We seem warranted, then, in assuming that the frictional resistance that must be overcome in an engine which is barely moving will be about four times as great as the initial friction of the engine at the piston speed at

which tests are usually made. We will assume, also, that the condition of the bearings, bearing surfaces and cylinder walls at the time the turning engine is used may be such as to increase the work of the engine by 50 percent.

The average initial friction is given by Taylor as being of such an amount as to require 2 pounds mean effective pressure upon the area of the low-pressure piston to overcome it.

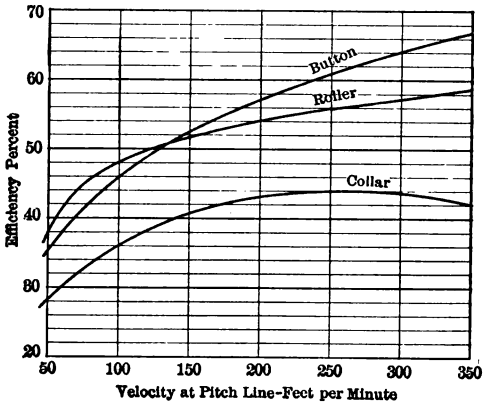


FIG. 4.

Blechynden gives from 1.5 to 1.75 pounds mean effective pressure over the area of all the pistons as the force necessary. With the ordinary ratio of cylinder areas this gives from 2.5 to 2.75 pounds mean effective pressure referred to the low-pressure cylinder. The analysis of the *Yorktown* trial gave 1.6 pounds mean effective pressure referred to the low-pressure cylinder as the amount absorbed by initial friction; in the *Kearsarge* it was 1.95 pounds, and in the *Massachusetts* 1.5 pounds.

All things considered, it would seem wise to assume that the mean referred pressure necessary to overcome the initial friction will vary from 2 pounds in naval and other lightly-

built engines, where the parts are small relative to the power developed, to 2.5 pounds in the ordinary type of triple engines for merchant ships, and up to 3 pounds for six-cylinder quadruple engines.

EFFICIENCY OF WORM GEARING.

The efficiency of worm gearing varies considerably, depending upon the care taken in its design. Of course, as much care need not be taken in the case of gears for a turning engine, which is used but seldom, and then only for a few minutes at a time, as would be necessary in the case of gears for machine tools. Nevertheless, it is well to bear in mind the principles upon which good design depends. First, the angle of the thread should be as large as possible up to 30 degrees or more for worm gearing, where the maximum efficiency is obtained. With spiral gears the maximum efficiency occurs at an angle of about 45 degrees, but the shafts are not at right angles, so there is not as much side thrust upon the spiral gears as there would be in the case of a worm and wheel with this angle of thread. In order to get these large angles it is customary to use double, triple and quadruple threads. Single threads are always used in turning gears, so the thread angle will not be greater than 9 degrees or 10 degrees, and is more often less than 7 degrees.

Second, the efficiency increases with the velocity at the pitch circle of the worm up to a speed of about 250 feet per minute.

Third, in order to prevent cutting and keep the worm cool, the thrust upon the teeth should decrease as the velocity at the pitch circle increases.

Fig. 3 gives the results of experiments made by Mr. Lewis for William Sellers Company. These curves seem to be in accord with the results obtained from practice (see *Worm and Spiral Gearing,* by F. A. Halsey). These curves show the advantage of using large pitch angles. The efficiencies shown by Fig. 3 can be obtained if the worm is run in an oil bath, and

if some care is taken in the design of the thrust bearing for the worm.

Fig. 4 shows the results of tests by Prof. Thurston upon three worms. In the one marked "collar," the thrust was taken by the end of the shaft; in the one marked "button," it was taken by the point of a set screw; in the one marked "roller," it was taken by ball bearings. The lower curve represents the conditions that one is likely to meet in the case of worms for turning gears, so it would appear that an efficiency of .4 is about all that will usually be obtained. The lost work, .6 of the total work, will be used up in overcoming friction between the teeth, friction in the thrust bearing of the worm, and friction in the bearing of the worm wheel due to the force acting upon the teeth, at the pitch circle and to the side thrust of the worm. We will assume that the loss in each of the above places mentioned is .2 of the total power driving the worm. In figuring the teeth, then, we shall assume that .6 of the power driving the worm passes through them.

Experiments have been made to determine the relation between the speed at the pitch line of the worm and the force acting normal to the face of the teeth when cutting begins. Lewis found that with a velocity of sliding of 306 feet per minute, a force of 5,558 pounds produced no cutting and no change in efficiency. With a speed of 360 feet per minute and a force of 4,837 pounds, cutting commenced after running six minutes. With a velocity of 400 feet per minute and a force of 3,481 pounds, cutting commenced after three minutes. The products of these quantities are as follows:

Velocity of sliding, feet per minute	306	360	400
Force × velocity	1,700,748	1,741,320	1,392,400

The experiments made by C. Bach and E. Roser, republished in the *American Machinist,* July 16 and 23, 1903, show smaller values of "force × velocity," but the curves given show that

the value of this product increases as the velocity decreases.

The mechanical efficiency of the turning engine will probably be not less than .8, and if the cut-off in the cylinder is about .7 of the stroke the mean effective pressure will be about .75 of the initial pressure absolute. The turning engine takes steam from the auxiliary steam line, where the pressure will probably not be more than 100 pounds, and very often it will be less than that. It is advisable to put in a large enough cylinder so that the engine will run with an initial pressure in the cylinder of 50 pounds gage, or 65 pounds absolute. The stroke of the turning engine is usually from 6 to 8 inches, and the diameter also is usually within this range. When the conditions are such as to call for more than 8 inches diameter, it is advisable to use two cylinders.

Fig. 3 shows that the efficiency of worm gearing increases as the pitch angle becomes larger. In order that the pitch angle of the worm may be as large as possible, it is best to keep the diameter as small as possible. If the small worm is cut out of the turning engine crank shaft forging, the diameter of its pitch cylinder can be about 2.5 times the pitch of the teeth. If it is a separate piece keyed to the shaft, the pitch cylinder will have a diameter of about 3 times the pitch of the teeth, in order that the thickness of metal below the root of the thread may be from .5 to .6 times the pitch of the teeth. The length of the worm should be from 3 to 4 times the pitch of the teeth.

If all of our assumptions are correct, we find that the frictional work to be overcome in one revolution of the engine is

$$\frac{\pi D^2}{4} \times MEP_t \times f \times 2S \times g \qquad (2)$$

Where D = diameter of low-pressure cylinder of main engine in inches.

MEP_t = frictional mean effective pressure referred to the low-pressure cylinder.
= 2 for naval and other light engines.
= 2.5 for engines of merchant ships.
= 3 for six-cylinder quadruple engines.
f = ratio between coefficients of friction of rest and motion = 4.
S = stroke of main engine in inches.
g = coefficient to allow for worst condition of bearings and surfaces = 1.5.

POWER OF TURNING ENGINE.

The power of the turning engine must be such that the work delivered to the crank shaft of the main engine shall equal (2). The quantities involved in the development and transmission of the power are as follows:

$$\frac{\pi d^2}{4} \times mep \times 2s \times n \times n_1 \times e \times e_1 \times e_2 \qquad (3)$$

where d = diameter of turning engine cylinder in inches.
mep = mean effective pressure upon the piston of the turning engine.
s = stroke of turning engine in inches.
n = number of teeth on small worm wheel.
n_1 = number of teeth on large worm wheel.
e = efficiency of small worm wheel.
e_1 = efficiency of large worm wheel.
e_2 = efficiency of turning engine.

This can be put into the form:

$$\frac{\pi d^2}{4} \times .75P \times 2s \times n \times n_1 \times .4 \times .4 \times .8 \qquad (4)$$

And when equated to (2) can be reduced to the form:

$$d^2 s = \frac{D^2 \times MEP_t \times S}{n \times n_1 \times P} \times 63.0 \qquad (5)$$

where P = initial pressure absolute at turning engine.

The force acting upon the teeth of the small worm wheel at the pitch circle will be:

$$F = .75P \times \frac{\pi d^2}{4} \times 2s \times .8 \times \frac{1}{p} \times .6 \qquad (6)$$

Where p = pitch of teeth on small worm in inches,

$$\text{Let } Z = .75P \times \frac{\pi d^2}{4} \times s \qquad (7)$$

$$\text{Then } F = .96Z \times \frac{1}{p} \qquad (8)$$

The force acting upon the teeth of the large worm wheel at the pitch circle will be:

$$F_1 = \frac{2}{3} F \times \frac{p \times n}{p_1} \times .6 \qquad (9)$$

where p_1 = pitch of teeth on large worm in inches.

$$\text{Therefore } F_1 = .384\ Z \times \frac{n}{p_1} \qquad (10)$$

The turning moment in inch pounds upon the main engine crank shaft will be:

$$M_1 = \frac{2}{3} F_1 \times \frac{n_1 p_1}{2\pi} = .041\ Z\ n\ n_1 \qquad (11)$$

The turning moment in inch pounds upon the shaft of the large worm will be:

$$M = F \times \frac{n p}{2\pi} = .153\ Z\ n \qquad (12)$$

In the case of the large worm wheel whose diameter is limited by the height of the engine bed:

$$p_1 = \frac{C \pi S}{n_1} \qquad (13)$$

where C is a coefficient whose value varies from 1.1 to 1.5.

Therefore

$$F_1 = .384\ Z \times n \times n_1 \times \frac{1}{C \pi S} = .122\ Z \times n \times n_1 \times \frac{1}{C S} \qquad (14)$$

Since Z is a function of Pd^2s, it will be seen by referring to equation (5) that for a given engine Znn_1 is a fixed quantity, hence the force acting upon the teeth of the large wheel can be reduced only by increasing C.

If we assume that the force at the pitch circle is taken by two teeth, the stress at the root of the teeth of the small worm wheel will be:

$$f = \frac{F\ l\ 6}{2d\ t^2} = \frac{4.2\ F}{C_1\ p^2} \qquad (15)$$

Where l = length of teeth below pitch circle $= .35\ p$.

b = breadth of teeth at root $= C_1\ p$ (C_1 varies from 2 to 2.5).

t = thickness of teeth at root, assumed to be $.5\ p$ for teeth whose thickness at the pitch circle $= .48\ p$.

The stress at the root of the teeth of the large worm wheel will be:

$$f_1 = \frac{F_1\ l_1\ 6}{2b_1\ t_1^2} = \frac{2.91\ F_1}{C_1\ p_1^2} \qquad (16)$$

$l_1 = .35 p_1$, $b_1 = C_1\ p_1$ where C_1 varies from 2. — 2.5.

t_1 is assumed to be $.6\ p_1$ when thickness of teeth at pitch circle $= .48\ p_1$.

To determine the number of teeth n_1 on the large worm wheel, we have from equation (16):

$$f_1 = \frac{2.91\ F_1}{c_1\ p_1^2}, \text{ but } F_1 = \frac{.122\ Z\ n\ n_1}{C S}, \text{ and } p_1 = \frac{C \pi S}{n_1}$$

Therefore

$$f_1 = \frac{Z\, n\, n_1^3}{27.7\, C_1\, C^3\, S^3}, \text{ or } n\, n_1^3 = \frac{27.7\, f_1\, C_1\, C^3\, S^3}{Z}$$

$$n\, n_1 = r\, m$$

Therefore

$$n_1^3 = \frac{27.7\, f_1\, C_1\, C^3\, S^3}{Z\, r\, m} \qquad (17)$$

$$n = \frac{r\, m}{n_1}$$

n should not be less than 25, preferably 30.

Referring to equation (5), it will be seen that the denominator of equation (17) is a fixed quantity for a given engine [see also equations (1) and (7)], so any variation in the number of teeth, and consequently in the pitch of the large worm wheel, will have to be made by varying the stress f_1, the tooth breadth factor C_1, and the wheel diameter factor C. In short-stroke engines, where the diameter of the large worm wheel will have to be small on account of the lack of space between the shaft and inner bottom, the wheel will be rather broad and of large pitch. If possible, it is well to have the stress at the root of the teeth not more than 3,500 for cast iron and 5,000 for cast steel. In short-stroke engines the value of f_1 can be 4,000, and wheels have been designed with $f_1 = 4{,}500$. It is advisable to make the worm wheels of cast iron and the worms of steel, as these work well together and the worm has more wear on it than the wheel. If the teeth of the worm and wheel are of equal thickness at the pitch circle, the stress at the root of the worm teeth does not need to be figured, as the worm teeth will be thicker than those of the wheel. When the teeth are of unequal thickness, the stress at the root of the worm teeth should be figured also. When the wheel is made of cast steel, the worm should be of bronze, as cast steel and wrought steel do not work well together.

CALCULATIONS.

We will design a turning gear for the engine whose calculations were carried through by the author in the previous chapters on "Marine Engine Design."

Indicated horsepower = 3,000. Piston speed = 850 feet per minute. Boiler pressure = 185 gage. Cylinder sizes and stroke, 23½ inches, 41 inches, 64 inches by 42 inches. Let the main engine be turned over once in eight minutes, the turning engine making 250 revolutions per minute. Then, from equation (1), $n\, n_1 = 8 \times 250 = 2{,}000$.

From equation (5), $d^2 s = \dfrac{(64)^2 \times 2.5 \times 42}{2{,}000 \times 65} = 208.$

Let $s = .6$ inches, then $d = 5.9$ inches; use 6 inches.

The mean effective pressure of the turning engine will be $.75 \times 65. = 49$ pounds, and the area of the cylinder 28.3 square inches.

$Z = 49 \times 28.3 \times 6 = 8{,}325$-inch pounds.

We will make the worm wheel of cast iron and the worms of mild steel. Let the stress at the root of the teeth be not more than 4,000 pounds in the large worm wheel, the breadth of the wheel not more than 2.5 p_1 and the diameter of the pitch circle 1.5 S. From equation (17),

$$n_1^2 = \frac{27.7 \times 4{,}000 \times 2.5 \times (1.5)^3 \times (42)^2}{8{,}325 \times 2{,}000} = 4{,}145.$$

$n_1 = 64.4$; use 64 teeth.

$$n = \frac{2{,}000}{64} = 31.2; \text{ use 31 teeth.}$$

From equation (13),

$$p_1 = \frac{1.5 \pi 42}{64} = 3.085 \text{ inches; use 3.25 inches and}$$

reduce the breadth of wheel.

From equation (10),

$$F_1 = \frac{.384 \times 8{,}325 \times 31}{3.25} = 30{,}550.$$

From equation (16),

$$C_1 = \frac{2.91 \times 30{,}550}{4{,}000 \times (3.25)^2} = 2.11.$$

$2.11 \times 3.25 = 6.87$ inches; use 7 inches breadth of wheel.

From equation (16),

$$f_1 = \frac{2.91 \times 30{,}550}{2.15 \times (3.25)^2} = 3{,}920 \text{ pounds.}$$

Diameter of pitch circle of wheel $= \frac{64 \times 3.25}{\pi} = 66.2$ inches.

For the small worm wheel we will use a pitch of 1.75 inches and let the breadth be $2.0 \times 1.75 = 3.5$ inches.

Then from equation (8),

$$F = \frac{.96 \times 8{,}325}{1.75} = 4.570 \text{ pounds.}$$

Then from equation (15),

$$f = \frac{4.2 \times 4{,}570}{2.0 \times (1.75)^2} = 3{,}140 \text{ pounds per square inch.}$$

This wheel can be made narrower if it is desirable, as the stress is low, or the pitch could be slightly reduced. The diameter of the small worm wheel will be:

$$\frac{1.75 \times 31}{\pi} = 17.28 \text{ inches.}$$

The indicated horsepower of the turning engine will be:

$$\frac{2 \times 49 \times .5 \times 28.3 \times 250}{33{,}000} = 10.5 \text{ indicated horsepower.}$$

The mean twisting moment on the crank shaft of the turning engine will be:

$$\frac{63{,}000 \times 10.5}{250} = 2{,}650\text{-inch pounds.}$$

The maximum twisting moment will be $2 \times 2{,}650 = 5{,}300$-inch pounds.

On the assumption that the bending moment is equal to one-half the twisting moment, and that the allowable stress on the shaft is 7,500 pounds per square inch, we get for the diameter:

$$\text{Diameter} = 1.17 \times 1.72 \sqrt{\frac{5{,}300}{7{,}500}} = 1.79 \text{ ins.; use } 1.875 \text{ ins.}$$

The factor 1.17 was obtained from the following table given by Unwin:

Bending Moment.

Twisting Moment.

	.25	.50	.75	1.0	1.25	1.5	1.75	2.0	3.0
Factor—	1.09	1.17	1.26	1.34	1.42	1.49	1.56	1.62	1.83

The small worm is to be keyed to the shaft so the diameter at the root of the thread $= 1.875 + 1.1 \times 1.75 = 3.80$ inches; use 3.75 inches.

Length of tooth flank $= .35 \times 1.75 = .618$ inch; use .625 inch.

Length of face $= .3 \times 1.75 = .525$ inch; use .5 inch.

Diameter of pitch circle of worm $= 3.75$ inches $+ 1.25$ inches $= 5$ inches.

Extreme diameter of small worm $= 6$ inches.

$$\text{Angle of thread} = \tan^{-1} = \frac{1.75}{5\pi} = .1113 = 6.35 \text{ degrees.}$$

The crank shaft for the turning engine was designed to resist a twisting moment equal to twice the mean, and a bending moment equal to half the maximum twisting moment. The shaft of the large worm is not subject to such variations in

twisting moment, as the fly-wheel on the crank shaft will cause this moment to be more uniform. The ratio of bending moment to twisting moment will depend upon the breadth of the wheel, and the radius of its pitch circle. In this case the half breadth of the wheel is 3.5 inches and the radius about 8.5 inches, so the bending arm will be about .5 times the twisting arm. Since the stresses are more uniform, the allowable fibre stress can be increased to 9,000 pounds.

From (12), $M = .153 \times 8{,}325 \times 31 = 39{,}460$-inch pounds. Assume the maximum twisting moment $= 39{,}460 \times 1.25 = 49{,}325$-inch pounds. Diameter of shaft $=$

$$1.17 \times 1.72 \sqrt[3]{\frac{49{,}325}{9{,}000}} = 3.525 \text{ inches; use } 3.50 \text{ inches.}$$

Diameter of worm at root of thread $= 3.5 + 1.1 \times 3.25 =$ 7.075 inches; use 7 inches.

Length of flanks $= .35 \times 3.25$ inches $= 1.14$ inches; use 1.125 inches.

Length of faces $= .3 \times 3.25 = .975$ inch; use 1.0 inch.

Pitch diameter of worm $= 7$ inches $+ 2.25$ inches $= 9.25$ inches.

Outside diameter of worm $= 9.25$ inches $+ 2$ inches $= 11.25$ inches.

$$\text{Angle of thread} = \tan^{-1} = \frac{3.25}{9.25\,\pi} = .112 = 6.4 \text{ degrees.}$$

The velocity of the small worm at the pitch line will be:

$$\frac{250 \times \pi \times 5}{12} = 327.4 \text{ feet per minute.}$$

The force acting normal to the teeth is 4,570; force $\times$ velocity $= 4{,}570 \times 327.4 = 1{,}496{,}218$. Referring to Mr. Lewis' experiments, it will be seen that the worm will be likely to work without cutting for the short periods that the engine will run.

REVERSING ENGINES.

Reversing engines are required upon marine engines of any size. While the smaller engines may be reversed by hand, the attempt to do this in engines over 1,000 indicated horsepower would necessitate such a large multiplication of power that the operation would be too slow for safety.

Some of the types of reversing engines employed are shown in Figs. 1 to 4. There are two general types, the direct acting, shown in Figs. 1, 2 and 3, and the all round, shown in Fig. 4.

In Fig. 1, *A* is the steam cylinder, *B* is the oil-brake cylinder, *C* is the valve chest containing the valve which governs the admission of steam to *A*, and *D* is the valve chest which contains the valve governing the flow of oil from one side of the piston to the other. The oil cylinder is used to prevent the gear from getting too much momentum, and so doing damage when it is necessary to reverse quickly. It also causes the gear to move smoothly. Fig. 5 shows a section through the valve chest *C* of Fig. 1. In this case a piston valve is used for both steam and oil. The steam is taken in the middle of the valve and it exhausts at the ends. More commonly a slide valve is used, in which case the steam is taken at the ends of the valve and exhausts in the middle. The piston valve is so made that it overlaps the edges of each port by 1/16 or 3/32 inch. When no oil cylinder is used, as in the gear shown in Fig. 2, the exhaust lap is usually made larger, so that considerable compression may be obtained at the end of the stroke and produce the cushioning that the oil would. In the case of the valve for the oil cylinder the laps must be no more than are necessary for oil tightness, as the gear will be locked as soon as the valve closes.

In some cases the steam valve is made "line and line," or with no lap at all, and occasionally with a clearance. This is

done when it is thought necessary to have the steam acting upon the piston in the "ahead" position to keep the links from

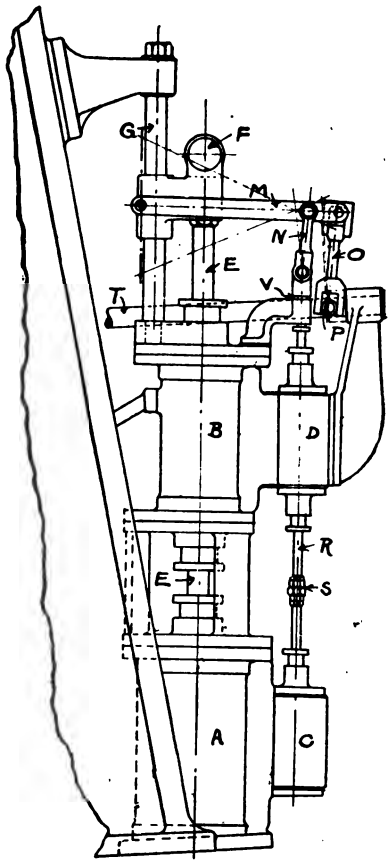


FIG. 1.

working over towards mid position. Referring to Fig. 5, it will be seen that if we attempt to make a negative steam lap on the valve there shown by lengthening or shortening the

valve stem with the adjusting nuts *S*, we shall increase the steam lap on the other end of the valve and throttle the steam when going from the "head" position to the "astern" position. In fact, it would be possible by this means to put so much clearance on the "ahead" position that the valve would not open the port to the other end of the cylinder.

The displacement of the valve is usually much less than the

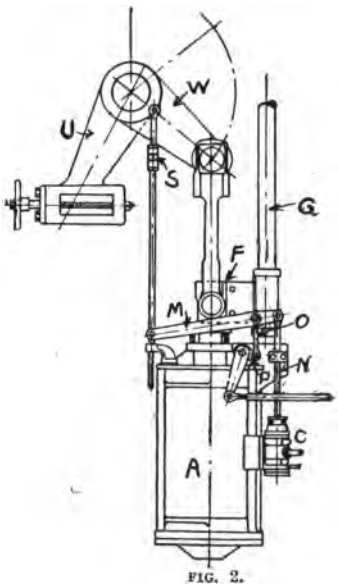


FIG. 2.

width of port. In Fig. 5 the lap of the valve is 3/32 inch, the width of the port is ⅞ inch, and the clearance is ¼ inch, so the maximum port opening of the valve is 5/32 inch. If the valve stem length were altered at *S*, Fig. 1, so that the lap on one end was increased from 1/32 to ¼ inch, the valve would not open at that end. This maximum port opening is made small so that the links will stop moving almost as soon as does the reverse lever. The greater the maximum port open-

ing the longer the links will keep moving after the reverse lever stops. The first movement of the reverse lever takes up the clearance at *V* in Fig. 1, and then it cannot move further until the crosshead commences to move. By following up the crosshead with the reverse lever the valve is kept open,

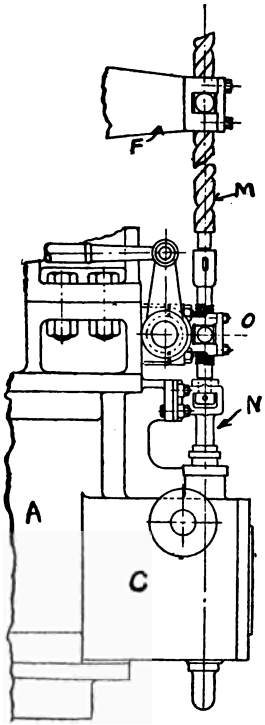


FIG. 3.

and then when the lever is stopped a very slight movement of the crosshead brings the valve back to its mid position. The link *O* is placed where it is most convenient, and the lengths of the bell-crank levers made such that the movement of *O* is to the movement of the crosshead as the distance of *O*

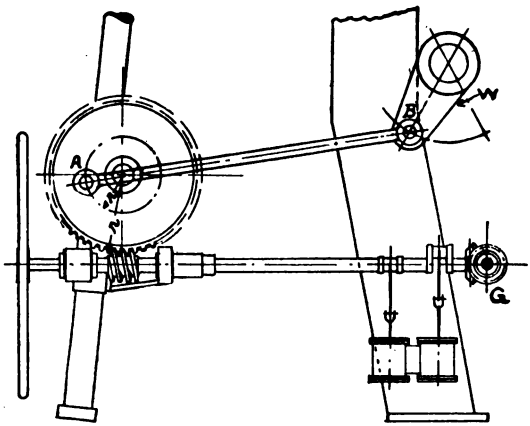


FIG. 4.

from the floating lever N is to the distance of the crosshead from that lever.

In the Brown gear shown in Fig. 3 the pitch of the coarse thread corresponds to the longer arm of the lever M in the other gears, and the pitch of the fine thread corresponds to

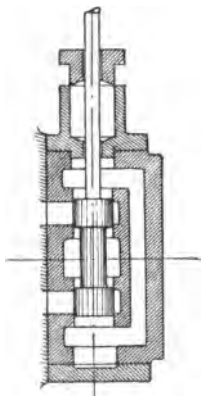


FIG. 5.

the shorter arm. The crosshead N corresponds to the floating lever N in the other gears. This type usually has an oil cylinder above the crosshead F, and the upper end of the valve spindle M is attached to the valve of the oil cylinder, and regulates the flow of oil from one side of the piston to the other. The diameter of the oil cylinder is usually made about two-thirds of the diameter of the steam cylinder.

It will be found usually that the diameter of the reversing engine cylinder is from .2 to .185 the diameter of the low-pressure cylinder of the main engine. It will vary with the boiler pressure, steam speeds and eccentricity. The reversing engine takes its steam from the main steam line between the boiler and the throttle valve, so as to have steam even when the main engine is shut down. The size of the piston should be such that with .9 the boiler pressure there will be a twisting moment exerted equal to twice that for which the reverse shaft was designed. If the steam speeds are low the piston valves will be larger in diameter and the slide valves will be broader, so the frictional work to be overcome in moving the valves will be greater. The greater the eccentricity the greater the distance through which the valves must be moved. Some consideration should therefore be given to these three conditions in settling upon the size of the reverse cylinder. The stroke of the reversing engine is usually about 2.5 the travel of the valves, but varies from two to three times this amount.

The reversing engine lever, or levers (see W, Fig. 2), in going from the ahead to the astern position must move through the same angle as do the reversing levers V, and the length of the former should be such that the angle between the extreme positions is subtended by a line whose length is equal to the stroke of the reversing engine. If Θ is this latter angle, and s is the stroke of the reversing engine, the length of the reversing engine lever will be:

$$l = \frac{s}{2 \sin \frac{1}{2}\theta} \qquad (1)$$

The piston rod can be calculated by the formula given for piston rods in the chapter on "Marine Engine Design."[1] The formula is as follows:

$$D^2 = \sqrt{\frac{0.48\ F\ C\ F}{10{,}000{,}000} + F^2} + F \qquad (2)$$

D = diameter of rod in parallel part.

$F = \frac{2\ W'}{\pi\ f}$

W = load on rod.

f = allowable stress per square inch = C/N.

C = ultimate strength of material = 60,000 — 75,000 pounds.

N = factor of safety — 15.

l = length of parallel body of rod in inches = stroke + diameter of cylinder.

The connecting rod can be calculated by the formula[2] given in the same article for connecting rods:

$$D^2 = \sqrt{\frac{1.08\ F\ C\ F}{10{,}000{,}000} + F^2} + F \qquad (3)$$

All the quantities are as above except that $N = 12$, and the length of the rod is usually from 1.5 to 2.5 the length of the reversing engine lever. This length must suit the position of the cylinder relative to the reverse shaft. When an oil cylinder is used and the crosshead is above the oil cylinder the rod will have the lower value given; when the crosshead is between the two cylinders, the length may exceed that given

[1] See page 49.

[2] See page 71.

above. The maximum force transmitted through the connecting rod will be slightly greater than that transmitted through the piston rod, due to the angle which the former makes with the center line of the engine at the middle and ends of the stroke. This increase is so slight, however, that it can be neglected and the rod calculated for the same load as the piston rod. The diameter of the rod at the ends can be reduced to .85 of the diameter at the middle.

The pressure upon the crosshead pin can be as high as 3,000 pounds per square inch, since the connecting rod moves through a very slight angle, and the engine will not be used with sufficient frequency to heat the pin.

The crosshead guide is usually a circular rod, and should be designed to resist the bending that will come upon it when the piston is at the middle of its stroke. The center line of the engine is so placed that the connecting rod will swing through equal angles on either side. The maximum load coming upon the guide will be:

$$w = \frac{r - \sqrt{r^2 - (s/2)^2}}{2 \times l} \times W \qquad (4)$$

r = length of the reversing engine lever.
s = stroke of reversing engine.
l = length of connecting rod.
W = load upon piston rod.

The pressure per square inch on the crosshead guide should be from 60 to 80 pounds.

In the All-round gear shown in Fig. 4, the radius r should be about 2.5 times the eccentricity of the valves, and the radius r of the pitch circle of the worm should be such that with the pitch used the number of teeth on the worm wheel will be between 30 and 45. A less number than 30 gives weak teeth and more than 45 makes the time of reversing rather long. With 30 to 45 teeth the reversing engine will have to make

from 15 to 23 revolutions to throw the links over if a single-threaded worm is used. The design of worms and worm wheels has been taken up under the head of Turning Engines, and some of the formulæ there given for the small worm and worm wheel can be used here.

Referring to Fig. 4 it will be seen that

$$\frac{T}{l} = \frac{mep \times a \times 2s \times r \times e \times e_1}{p\ r_1} \qquad (5)$$

and if $a = \frac{\pi d^2}{4}$, and $p = \frac{2\pi r}{n}$,

$$d^2 s = \frac{4\,T\,r_1}{l \times mep \times n \times e \times e_1} \qquad (6)$$

T = twisting moment in inch pounds on reverse shaft necessary to throw all the gears.
l = length in inches of reverse engine lever.
mep = mean effective pressure in cyliders of gear.
a = area of cylinder in square inches.
s = stroke of engine in inches.
r = radius of pitch circle of worm wheel in inches.
r_1 = eccentricity of pin A in inches. (See Fig. 4.)
p = pitch of teeth in inches.
n = number of teeth on worm wheel.
e = efficiency of worm.
e_1 = efficiency of engine.

Formula (15), page 151, gives the stress at the root of the teeth of the worm wheel:

$$f = \frac{4.2\ F}{C_1\ p^2}$$

Formula (8) in the same issue gives:

$$F = .96\ Z\ \frac{1}{p};$$

and from (7) in the same issue:

$$Z = mep \times a \times s.$$

Therefore, $$f = \frac{4.03\ mep \times a \times s}{C_1\ p^3} \qquad (7)$$

From formula (5):

$$mep \times a \times s = \frac{T\ p\ r_1}{2\ l\ r\ e\ e_1}$$

Therefore, $$f = \frac{2.02\ T\ r_1}{C_1\ p^3\ l\ r\ e\ e_1} \qquad (8)$$

There seems to be no reason why f should vary inversely as the efficiencies of the gear and of the engine. Referring to formula (6), page 150, it will be seen that F is a function of these efficiencies. The presence of e and e_1 in the denominator of (8) will therefore cause the expression to be independent of these quantities. Assuming that $e = .4$ and $e_1 = .8$, we have:

$$f = \frac{6.3\ T\ r_1}{C_1\ p^2\ l\ r} \qquad (9)$$

$$\text{Therefore, } p = \sqrt{\frac{6.3\ T\ r_1}{f\ C_1\ l\ r}} \qquad (10)$$

p, T, r_1 r and l are as above.

f = allowable stress at root of teeth.

= 3,500-4,000 for cast iron and 5,000 for cast steel.

C_1 = breadth of wheel at root circle in terms of the pitch. C_1 is usually about 2.

CALCULATIONS.

In the chapter on "Marine Engine Design" the calculations were carried through for an engine working under the following conditions:

Indicated horsepower = 3,000.

Piston speed = 850 feet per minute.

Boiler pressure = 185 pounds gage.

Cylinder sizes and stroke, 23½ inches, 41 inches, 64 inches by 42 inches.

Eccentricity, 4.25 inches on the high and intermediate-pressure cylinders; 4.75 inches on the low-pressure.

The reversing engine was placed between the low-pressure and intermediate-pressure gears, and the twisting moment from the low-pressure gear was 210,000 inch pounds. This was obtained by taking into consideration the weight, inertia and friction of the low-pressure valves. It was assumed that if the engine were in such a position that the maximum power had to be exerted to reverse the low-pressure gear the other two gears would be in a better position, and only the same amount of power would be necessary to reverse both the high-pressure and medium-pressure gears.

The twisting moment that the reverse engine cylinder must exert will then be 420,000 inch pounds. We will take the stroke of the engine to be

$$2.5 \times 8.5 = 21.25; \text{ use } 21 \text{ inches.}$$

Since the reverse levers were designed to move through an angle of 75 degrees, the length of the reversing engine lever will be, by formula (1):

$$l = \frac{21}{2 \times .61} = 17.2; \text{ use } 17.25 \text{ inches.}$$

$$\frac{420{,}000}{17.25} = 24{,}350 \text{ pounds,}$$

the force that must act upon the pin of the reversing engine lever. Assume the back pressure in the cylinder to be 10 pounds absolute:

$$200 \times .9 - 10 = 170 \text{ pounds per square inch.}$$

$$\frac{24{,}350}{170} = 143.3 \text{ square inches} = \text{area of piston.}$$

A diameter of 13.53 inches will give this area; use a 14-inch cylinder. The reversing engine will be 14 inches by 21 inches. Since the diameter of the low-pressure cylinder is 64 inches this makes the cylinder of the reversing engine .219 times the low-pressure diameter.

PISTON ROD.

The length of rod will be about $21 + 14 = 35$ inches.

By formula (2):

$$F = \frac{2 \times 24{,}350}{\times\ 4{,}000} = 3.88$$

$$D^2 = \sqrt{\frac{.48 \times 3.88 \times 60{,}000 \times 35^2}{10{,}000{,}000}} + 15.05 + 3.88$$

$D = 3.04$; use 3 inches.

CONNECTING ROD.

Since no oil cylinder is to be used on this gear the length of the connecting rod can be about

$$17.25 \times 2.5 = 43.4; \text{ use } 42 \text{ inches.}$$

By formula (3):

$$F = \frac{2 \times 24{,}350}{\pi \times 5{,}000} = 3.1$$

$$D^2 = \sqrt{\frac{1.c8 \times 3.1 \times 60{,}000 \times 42^2}{10{,}000{,}000}} + 9.6 + 3.1$$

$D = 3.14$; use 3.25 inches.

Diameter at ends $= 3.25 \times .85 = 2.76$; use 2.75 inches.

CROSSHEAD GUIDE.

By formula (4):

$$w = \frac{17.25 - \sqrt{17.25^2 - 10.5^2}}{2 \times 42} \times 24{,}350 = 1{,}030 \text{ pounds.}$$

The length of guide from center to center of supports will be about the same as the length of the piston rod.

$$M = \frac{W\,l}{4} = \frac{1{,}030 \times 35}{4} = 9{,}000$$

For a beam of round section $D = \sqrt[3]{\frac{M\ 10.2}{f}}$

$$D = \sqrt[3]{\frac{9{,}000 \times 10.2}{5{,}000}} = 2.64;\ \text{use } 2.75 \text{ inches diameter.}$$

$$\text{Length of slipper} = \frac{1{,}030}{2.75 \times 70} = 5.36;\ \text{use } 5.5 \text{ inches.}$$

$$\text{Surface of crosshead pin} = \frac{24{,}350}{3{,}000} = 8.12 \text{ square inches.}$$

Length of bearing surface of pin = diameter of connecting rod at upper end = 2.75 inches:

$$\frac{8.12}{2.75} = 2.95;\ \text{use } 3 \text{ inches diameter.}$$

If the All-round type of gear is used the proportions will be as given below. Assume r to be 19 inches.

From formula (10):

$$p = \sqrt{\frac{6.3 \times 420{,}000 \times 10.5}{4{,}000 \times 2 \times 17.25 \times 19}} = 3.22;\ \text{use } 3.25 \text{ inches.}$$

$$n = \frac{2\ \pi\ 10}{3.25} = 36.7;\ \text{use } 37 \text{ inches.}$$

$$r = \frac{37 \times 3.25}{2\ \pi} = 19.15 \text{ inches.}$$

We will use steam at boiler pressure for the engine, and assume the m. e. p. to be .8 the absolute pressure; m. e. p. = $200 \times .8 = 160$ pounds.

From formula (6):

$$d^2 s = \frac{4 \times 420{,}000 \times 10.5}{17.25 \times 160 \times 37 \times .4 \times .8} = 540$$

Let $s = 6$ inches, $d^2 = 90$ inches. Use two cylinders 6.75 inches diameter by 6-inch stroke.

The proportions of the worm, worm wheel and shaft can be found in the same way as those for the turning engine. The length of the connecting rod will be determined by the position of the worm wheel. The size of the connecting rod and of the pins can be determined by the rules already given.

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